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CRYOGENIC PROPELLANT DENSIFICATION STUDY

Final Report

November 1978

by R. O. Ewart and R. H. Dergance

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by R. O. Ewart and R. H. Dergance

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Page i, 16. Abstract, line 6: The value 8.2×10^5 Kg (1.8×10^6 lb) should be 141,808 Kg (312,630 lb).

Page 6, Figure II-1, left-hand figure: The values

61.9 (203)
202,753 (446,993)
207,624 (457,734)
1,453,373 (3,204,139)
1,924,674 (4,243,136)

should be

54.1 (177.4)
120,785 (266,282)
135,871 (299,542)
951,098 (2,096,791)
1,259,066 (2,775,736)

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16. Abstract A study of ground and vehicle system requirements for the use of densified cryogenic propellants in advanced space transportation systems was conducted. Propellants studied were slush and triple point liquid hydrogen, triple point liquid oxygen, and slush and triple point liquid methane. Areas of study included propellant production, storage, transfer, vehicle loading and system requirements definition. A savings of approximately 8.2×10^5 Kg (1.8×10^6 lb) can be achieved in single stage to orbit (SSTO) gross liftoff weight (GLOW) for a payload of 29,484 Kg (65,000 lb) by utilizing densified cryogenics in place of normal boiling point propellants.					
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FOREWORD

This report was prepared by the Martin Marietta Corporation, Denver Division, under Contract NAS3-21014. The contract was administered by the Lewis Research Center of the National Aeronautics and Space Administration, Cleveland, Ohio. The study was performed from July 1977 to July 1978 and the NASA LeRC Project Manager was Mr. J. J. Notardonato.

The authors wish to acknowledge the contributions of the following individuals to this program: Mr. G. Presta and Mr. J. P. Gille for their thermal analyses; Mr. T. E. Bailey for the development of the storage vessel analytical models; and Dr. J. E. Anderson for the refinement of the model and system analyses.

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TABLE OF CONTENTS

	PAGE
LIST OF FIGURES	vi
LIST OF TABLES	x
SUMMARY	1
I. INTRODUCTION	3
II. DENSIFIED HYDROGEN DISCUSSION	5
A. Slush Hydrogen Storage and Transfer	5
1. Storage Tank	10
2. Transfer Line	20
3. Vehicle Tank	26
4. Total SLH ₂ System Transfer Analysis	28
B. Triple Point Liquid Hydrogen Storage and Transfer	44
1. Storage Tank	44
2. Transfer Line	50
3. Vehicle Tank	51
4. Total TP LH ₂ System Transfer Analysis	53
C. Production and Cost of Densified Hydrogen	66
1. Production	66
2. Cost of Densified Hydrogen	69
3. Location of Hydrogen Densification Plant	74
D. Densified Hydrogen Integrated Production and Loading System	78
1. System Component Description	82
2. System Operating Procedure	84
E. Other Densified Hydrogen System Considerations	86
1. System Safety	86
2. Slush Filtration	87
3. Slush Screens	87
F. Densified Hydrogen Conclusions	88
III. DENSIFIED OXYGEN DISCUSSION	91
A. Storage	93

TABLE OF CONTENTS (CONT)

	PAGE
B. Transfer	96
1. Storage Tank Pressurization	96
2. Pump and Transfer Line	98
C. Vehicle Tank Loading	100
D. Other TP LOX Considerations	104
1. Thermal Contraction Effect on LOX System	104
2. Cost of Triple Point Liquid Oxygen	106
E. Densified Oxygen System Conclusions	107
IV. DENSIFIED METHANE DISCUSSION	109
A. Slush Methane Storage and Transfer	109
1. SLCH ₄ Baseline System	109
2. Storage	111
3. Transfer	112
B. Triple Point Liquid Methane Storage and Transfer	115
1. TP LCH ₄ Baseline System	115
2. Storage	116
3. Transfer	117
C. Densified Methane Conclusions	118
V. OTHER DENSIFIED PROPELLANT CONSIDERATIONS	121
A. Ground System Instrumentation	121
B. Ground Transportation of Densified LH ₂ & LOX	122
VI. CONCLUSIONS	125
VII. RECOMMENDATIONS	127
APPENDIX A Cryogenic Tank Pressurization/Stratification Analysis Model	129
APPENDIX B Transfer Line Thermal Analysis	133
APPENDIX C SSTO Fuel Tank Insulation Analysis	149
APPENDIX D SSTO Oxidizer Tank Insulation Analysis	161
APPENDIX E SYMBOLS	167
REFERENCES	171
BIBLIOGRAPHY	175

LIST OF FIGURES

		PAGE
II-1	SSTO Vehicle Sizes Using Densified Propellants	6
II-2	KSC LC 39 SSTO/Densified Hydrogen Baseline Loading System	7
II-3	System Optimization Iteration Process	8
II-4	Slush Hydrogen Quality Decay in LC 39 LH ₂ Storage Tanks	11
II-5	SLH ₂ Storage Tank Pressurization for Varying Pressurant Flowrates	14
II-6	LC 39 SLH ₂ Storage Tank Heat Transfer to Slush from Pressurization	15
II-7	Storage Tank Stratified Layer Weight and Temperature at End of SSTO Loading vs. Line Size	18
II-8	SSTO SLH ₂ & TP LH ₂ Transfer Systems - Flowrate vs. Pressure Drop	21
II-9	SLH ₂ Quality Degradation in Ground Transfer System - Total Friction and Environmental Heating for Varying Size Transfer Lines	25
II-10	SLH ₂ & TP LH ₂ System Heat Input - SSTO Tank Heat vs. Loading Time	27
II-11	SSTO/SLH ₂ System Heat Input - Vehicle Tank and Transfer System Heat vs. Fast Fill Flowrate	30
II-12	SSTO/SLH ₂ System Heat Input - Total Heat vs. Fast Fill Flowrate	31
II-13	SSTO SLH ₂ Tank - Pad Hold Flowrate vs. Heat Flux	32
II-14	SSTO/SLH ₂ Propellant Loading Timeline - Storage Tank Profile for Existing 8 and 10 inch Transfer System	35
II-15	SSTO/SLH ₂ Propellant Loading Timeline - Vehicle Tank Profile for Existing 8 and 10 inch Transfer System	36
II-16	SSTO/SLH ₂ Propellant Loading Timeline - Storage Tank Profile for 12 inch Transfer System	37

LIST OF FIGURES (CONT)

		PAGE
II-17	SSTO/SLH ₂ Propellant Loading Timeline - Vehicle Tank Profile for 12 inch Transfer System	38
II-18	SLH ₂ Storage and Transfer Losses vs. SSTO Launch Rate	42
II-19	60% SLH ₂ Consumption vs. SSTO Launch Rate	43
II-20	Triple Point LH ₂ Density Decay in LC 39 LH ₂ Storage Tanks	45
II-21	Sub-Cooled LH ₂ Average Density Decay in LC 39 LH ₂ Storage Tanks	47
II-22	Sub-Cooled LH ₂ Average Temperature Rise in LC 39 LH ₂ Storage Tanks	48
II-23	TP LH ₂ Enthalpy Degradation vs. Flowrate - LC 39 Friction and Environmental Heating for Varying Transfer Line Sizes	52
II-24	SSTO/TP LH ₂ System Heat Input - Vehicle Tank & Transfer System Heat vs. Fast Fill Flowrate	54
II-25	SSTO/TP LH ₂ System Heat Input - Total Heat vs. Fast Flowrate for 1 inch PPO Foam Insulated Vehicle Tank	55
II-26	SSTO/TP LH ₂ Propellant Loading Timeline - Storage Tank Profile for Existing 8 and 10 inch Transfer System	58
II-27	SSTO/TP LH ₂ Propellant Loading Timeline - Vehicle Tank Profile for Existing 8 and 10 inch Transfer System	59
II-28	SSTO/TP LH ₂ Propellant Loading Timeline - Storage Tank Profile for 12 inch Transfer System	60
II-29	SSTO/TP LH ₂ Propellant Loading Timeline - Vehicle Tank Profile for 12 inch Transfer System	61
II-30	TP LH ₂ Storage and Transfer Losses vs. SSTO Launch Rate	64
II-31	TP LH ₂ Consumption vs. SSTO Launch Rate	65
II-32	Slush Hydrogen Production Facility Cost vs. Useful Life	70
II-33	Triple Point Liquid Hydrogen Production Facility Cost vs. Useful Life	71

LIST OF FIGURES (CONT)

		PAGE
II-34	Cost of Densified Liquid Hydrogen vs. Quality for Various Production Rates	73
II-35	LH ₂ Plant Location for SSTO - Cost vs. Quantity Delivered	77
II-36	Densified Hydrogen Integrated Production and Loading System for SSTO	79
III-1	KSC LC 39 SSTO/TP LOX Baseline Loading System	92
III-2	Triple Point LOX Density Decay in the LC 39 LOX Storage Tank	95
III-3	SSTO/TP LOX Propellant Loading Timeline Storage Tank Profile	99
III-4	SSTO Oxidizer Tank Configuration	103
III-5	SSTO/TP LOX Pad Hold Flowrate vs. Propellant ΔT for Varying Insulation Thickness	105
IV-1	SSTO/Densified Methane Baseline Loading System	110
A-1	Tank Pressurization/Stratification Model	130
B-1	Basic Transfer Line Cross-Section	134
B-2	Vacuum Jacketed Line Heat Leak Rate vs. Jacket Wall Temperature	136
B-3	Vacuum Jacketed Line Heat Leak Rate vs. Jacket Wall Temperature	137
B-4	Vacuum Jacketed Line with MLI Heat Leak Rate vs. Jacket Wall Temperature	139
B-5	Effect of Wall Temperature on Radiant Heat Transfer	142
C-1	SSTO Propellant Tank and Insulation System	150
C-2	SSTO SLH ₂ and TP LH ₂ Tank Insulation - Heat Flux vs. Thickness	154

LIST OF FIGURES (CONT)

		PAGE
C-3	SSTO SLH ₂ and TP LH ₂ Tank Insulation - Wall Temperature vs. Thickness	155
C-4	SSTO SLH ₂ and TP LH ₂ Tank Insulation - Total Heat Input vs. Thickness After 100 Minutes	156
C-5	SSTO/SLH ₂ and TP LH ₂ Tank Insulation - Gross Liftoff Weight vs. Thickness	157
C-6	SSTO SLH ₂ and TP LH ₂ Tank Insulation Thickness Cost Analysis	158
C-7	SSTO Total Program and Operations Cost versus Gross Liftoff Weight	159
D-1	Heat Transfer to Triple Point LOX vs. Thickness of Typical Spray Foam Insulation	164
D-2	Total Heat Gain of Triple Point LOX in SSTO Tanks vs. Time for Varying Insulation Thickness	165

LIST OF TABLES

		PAGE
II-1	LH ₂ Loading Sequence for Shuttle	8
II-2	SLH ₂ Storage Tank Heat Input	17
II-3	SLH ₂ Vehicle Tank Heat Input (1 inch PPO Foam)	28
II-4	SSTO/50% SLH ₂ Loading Sequence	34
II-5	SLH ₂ Storage and Transfer Refrigeration Loss	41
II-6	TP LH ₂ Storage Tank Heat Input	49
II-7	TP LH ₂ Vehicle Tank Heat Input (1 inch PPO Foam)	53
II-8	SSTO/TP LH ₂ Loading Sequence	57
II-9	TP LH ₂ Storage and Transfer Refrigeration Loss	63
II-10	Densified Hydrogen Production Costs	68
II-11	Cost of Densified Hydrogen	72
II-12	Cost of SLH ₂ Delivered to KSC from New Orleans	76
II-13	Cost to TP LH ₂ Delivered to KSC from New Orleans	76
II-14	Densified Hydrogen Integrated Production and Loading System for SSTO - Component Identification and Design Requirements	80
II-15	Gaseous Hydrogen Vented During Loading	83
III-1	LOX Loading Sequence for Shuttle	91
III-2	SSTO/TP LOX Baseline Loading Sequence	93
III-3	LC 39 LOX Transfer System Pressure Drop vs. Flowrate for NBP LOX and Atm TP LOX	97
III-4	LOX Pump Heat Input to Fluid	100
III-5	Pump and Transfer Line Heat Inputs	101

LIST OF TABLES (CONT)

		PAGE
III-6	Temperature and Density of Propellant Delivered to SSTO/TP LOX Tank	102
IV-1	SSTO/SLCH ₄ Baseline Loading Sequence	111
IV-2	LC 39 LH ₂ Transfer System Pressure Drop vs. Flowrate for 50% Slush Methane	113
IV-3	SLCH ₄ Slush Quality Change in Transfer Line	114
IV-4	SSTO TP LCH ₄ Baseline Loading Sequence	116
IV-5	LC 39 LH ₂ Transfer System Pressure Drop vs. Flowrate for TP Liquid Methane	117
IV-6	Average Temperature and Density Change of TP LCH ₄ in Transfer Line	118
B-1	Vacuum Jacket and Inner Line Dimensions	135
B-2	Gas Conduction Effect on Line Heat Leak	140
B-3	Basic Line Section Heat Leak Rates (BTU/hr-ft) - (530°R to 25°R)	141
B-4	Comparison of Line and Joint Heat Leak Rates	146
B-5	KSC Transfer System Components and Heat Leakage Rates	147
B-6	LH ₂ and LOX Line Heat Leak Rates	148

SUMMARY

Densified cryogenic propellants offer the advantage of smaller, lighter weight vehicle designs for space travel compared to normal boiling point cryogenics since the same propellant can be stored in a reduced volume tank. Therefore, these propellants are being considered for advanced space transportation systems to enhance vehicle payload capability as a function of gross lift-off weight. The densified propellants which offer the greatest potential reduction in vehicle size are slush and triple point liquid hydrogen, triple point liquid oxygen, and slush and triple point liquid methane.

This study was conducted to determine the feasibility of utilizing these propellants for launch vehicles from a ground system standpoint and to establish basic production, storage, transfer and vehicle loading design requirements. The basis for the study was to determine the adequacy of the existing Space Shuttle ground systems at Kennedy Space Center to support a single-stage-to-orbit (SSTO) vehicle loading and to identify required system modifications.

The densified hydrogen analyses were more extensive since considerably more experimental data are available in this area. In addition, NASA and industry interest was directed toward densified hydrogen for Shuttle application during the course of this study. The optimum system design and loading sequence for slush and triple point liquid hydrogen were established. The advantage of using slush over triple point liquid was clearly shown to the extent that future considerations should be directed toward slush alone.

The feasibility of using triple point liquid oxygen was shown with certain reservations and the addition of a recirculation system for upgrading and pad-hold loading operations.

The analyses of densified methane revealed that a system of a design similar to the existing liquid hydrogen ground system could be used for loading the SSTO vehicle.

I. INTRODUCTION

The Space Shuttle provides space transportation capabilities into the 1990's with the next generation space transportation system expected to be a totally reusable vehicle such as the Single Stage to Orbit (SSTO) with development scheduled before 1995. To achieve the goal of total reusability while maximizing SSTO performance significant advancements in propulsion system technology must be realized.

Liquid hydrogen and liquid oxygen have been used successfully as propellants for many launch vehicles because of their relatively high specific impulse and high specific energy. A significant disadvantage of these propellants is their low densities which produce penalties in vehicle gross liftoff weight (GLOW) due to the size requirements of the propellant tankage. The low heat capacities per unit volume of these cryogenics are also a key disadvantage in that boiloff losses and propellant quality maintenance prior to launch introduce undesirable operational complexity and storage capacity in the ground handling equipment and procedures. These two properties can be improved by subcooling or solidifying all or part of the liquid. For example, if liquid hydrogen is subcooled at its triple point temperature of 13.8°K (24.9°R) from normal boiling point (NBP) of 20.3°K (36.5°R), its density increases 8.8% and its capacity to absorb heat is increased by 20%. If cooling continues at the triple point temperature until 60% of the total mass is solid (60% slush hydrogen), a 16.8% density increase and a 34% heat capacity increase over normal boiling point liquid results. The increased density allows for the storage of the same mass of propellant in a smaller volume, thus reducing tank size and overall vehicle weight. The increased heat capacity allows for more heat to be absorbed before vaporization occurs.

Because of the advantages of subcooled (densified) propellants compared to NBP liquids their characteristics have been under investigation for several years, primarily at the National Bureau of Standards, Boulder, Colorado, with the major emphasis directed toward triple point and slush hydrogen. Through laboratory scale testing, physical properties, method of production, transfer and pumping losses, mixing, aging effects and instrumentation requirements relative to densified LH_2 have been investigated.

Current technology interest centers on slush and triple point (TP) liquid hydrogen and TP liquid oxygen. The use of slush instead of TP LOX represents a density increase of approximately 2%. However, the oxygen tank is small (33% of the volume of LH_2 tank at an oxidizer to fuel Mixture Ratio of 6.0) and does not drive the vehicle design and resultant GLOW. Therefore, slush LOX is not considered a viable candidate for technology advancement.

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Slush and triple point methane have been considered as potential first stage fuels in a dual mode SSTO configuration due to significant density advantages [482 Kg/m³ (30.1 lb/ft³)] when compared with LH₂ [70.8 Kg/m³ (4.42 lb/ft³)].

The use of these densified cryogens as propeplants for advanced space transportation systems will require that large quantities be produced, stored and transferred to support the planned mission models. The intent of this study, therefore, is to evaluate and define the large scale systems necessary to utilize these propellants of interest and to identify areas where further analytical and experimental studies are required.

This study deals primarily with the systems required to produce, store and load the space vehicle for launch and not the airborne propulsion system. This includes the production plant, ground storage tanks, transfer system and flight tank. The baseline system for this study consists of the SSTO vehicle and the cryogenic transfer systems at Kennedy Space Center Launch Complex 39 (KSC LC 39) presently being modified for Shuttle.

The method of analysis was one of a parametric evaluation of the system components (storage tank, transfer line, vehicle tank) followed by several overall system iterations to define the optimum loading sequence, flowrates and propellant consumed per launch. Also, methods of production, plant location, propellant cost, instrumentation and safety requirements were evaluated.

Finally, a system for the production and loading the SSTO with densified propellant was defined and the economic advantages evaluated.

To maximize the output of this study, the analyses were performed using English units. The text and the majority of the figures and tables are presented in SI as well as English units. Where conversion to SI units would reduce the usefulness of the report for its primary recipients only English units are presented.

II. DENSIFIED HYDROGEN DISCUSSION

A. SLUSH HYDROGEN STORAGE AND TRANSFER

The slush liquid hydrogen (SLH₂) analyses were conducted with a set of baseline requirements which included an SSTO vehicle (References 1, 2 and 3) utilizing 50% SLH₂ and triple point LOX at an MR of 6:1 as propellants (see Figure II-1). This vehicle has a gross liftoff weight (GLOW) of 1,117,246 Kg (2,463,106 lb) and a SLH₂ capacity of 120,510 Kg (265,679 lb) or 1,474.9 m³ (389,633 gal) at a density of 82 Kg/m³ (5.1 lb/ft³). For comparison, the fully loaded Space Shuttle external tank contains 102,283 Kg (225,495 lb) of liquid hydrogen at a density of 70.8 Kg/m³ (4.42 lb/ft³). The payload capacity of the SSTO is 29,485 Kg (65,000 lb) to low earth orbit, which is the same as the goal for Shuttle.

The existing Shuttle LH₂ loading system at KSC (References 4, 5 and 6) is used as the baseline for the ground system analysis. The major components of the system are an 3217.6 m³ (850,000 gal) (LH₂) capacity vacuum jacketed storage dewar; an ambient temperature vaporizer for storage tank pressurization; approximately 520 m (1,700 ft) of 25 cm (10-inch) diameter and 15 m (50 ft) of 20 cm (8-inch) diameter vacuum jacketed multilayer insulated transfer line; vent lines; and a burn pond for vent gas disposal. This system is shown schematically in Figure II-2 and represents the densified hydrogen baseline loading system for SSTO.

For Shuttle, the transfer of LH₂ is accomplished by pressurizing the storage tank to 550 KPa (65² psig) with gaseous hydrogen which is generated by vaporizing liquid from the tank. This is the pressure required to overcome the line and component pressure drops plus the vehicle tank head pressure. It is important to note that the fuel tank is located aft of the oxidizer tank in the Shuttle external tank (ET) and forward of the oxidizer tank in the SSTO. This results in an increase in head elevation of 26.2 m (86 ft) for the SSTO system. Flowrates to the vehicle are controlled by appropriate position of the transfer, chilldown and replenish valves.

The LH₂ loading sequence for Shuttle (Reference 7) is shown in Table II-1, and serves as a basis for the loading timeline analysis.

The remainder of this section is devoted to defining the optimum system design and loading sequence that will fill the SSTO vehicle with 50% SLH₂ at liftoff and, also, determine if the existing LC 39 LH₂ system is capable of loading SLH₂. To determine the optimum system and sequence, the thermal effects of each component part of

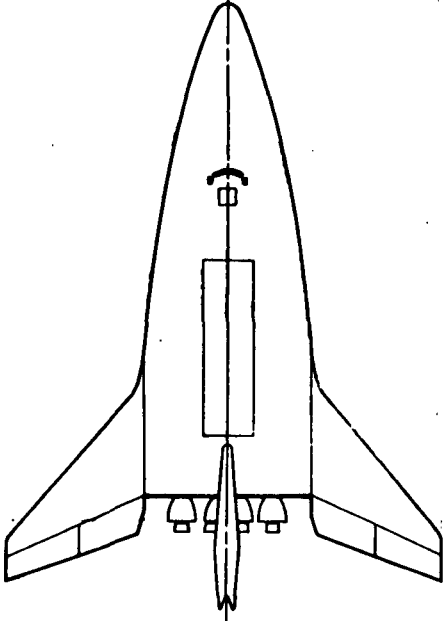
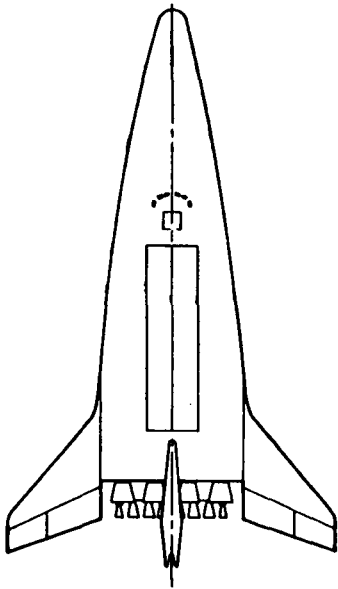
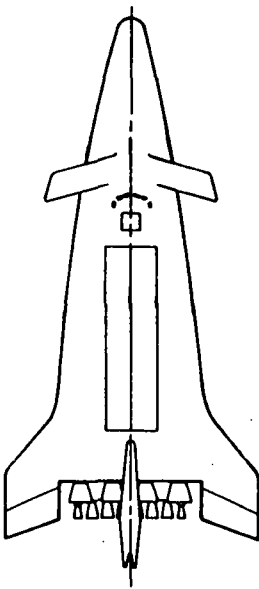
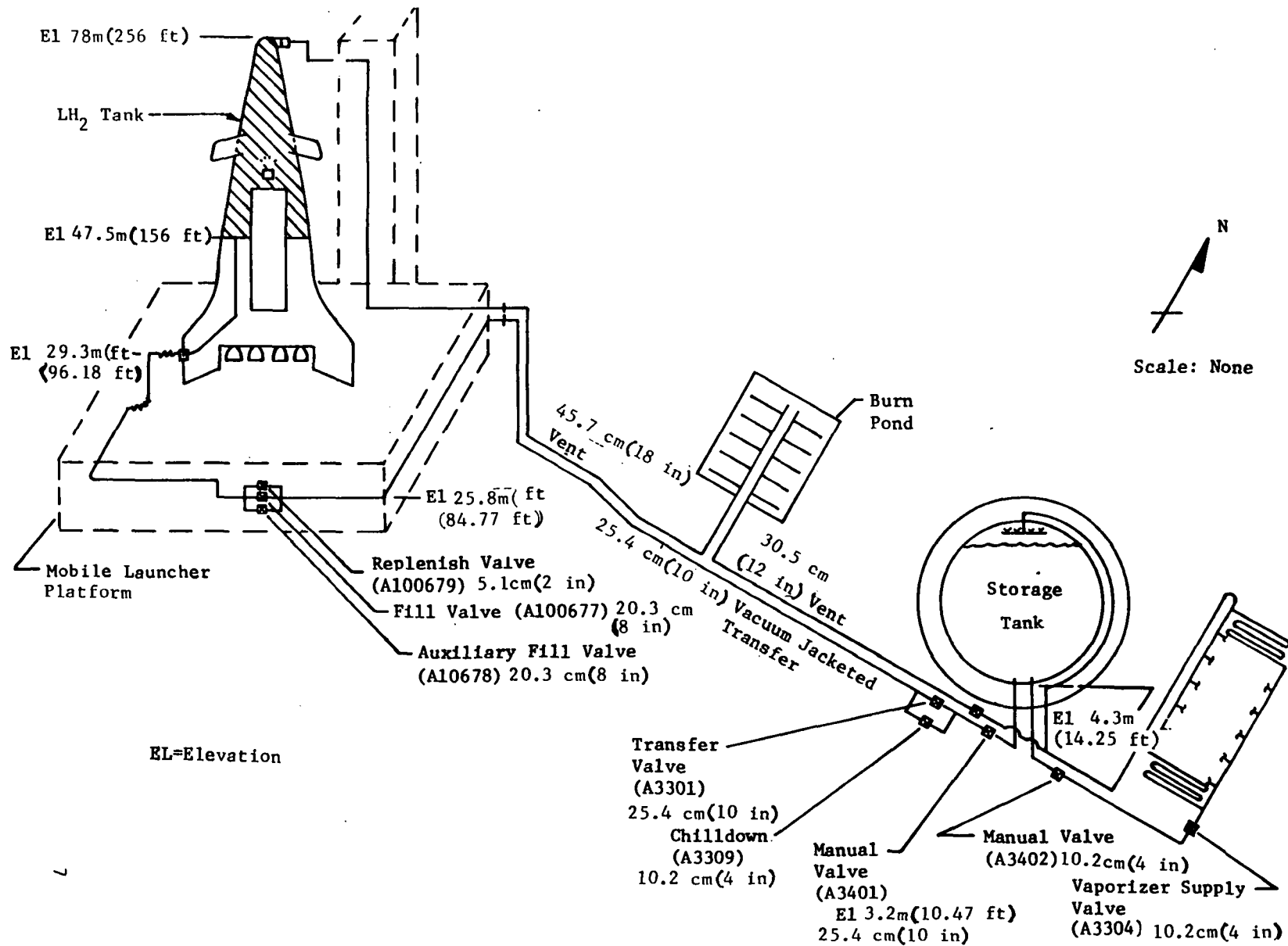
NBP LH ₂ and LOX Propulsion	TP LH ₂ and TP LOX Propulsion	50% SLH ₂ and TP LOX Propulsion
		
Length, m(ft) 61.9(203) Dry Weight, Kg(lb) 202,753(446,993) LH ₂ Weight, Kg(lb) 207,624(457,734) LOX Weight, Kg(lb) 1,453,373(3,204,139) Gross Liftoff Weight, Kg(lb) 1,924,674(4,243,136)	Length, m(ft) 50.3(164.9) Dry Weight, Kg(lb) 107,322(236,601) LH ₂ Weight, Kg(lb) 123,657(272,615) LOX Weight, Kg(lb) 865,612(1,908,306) Gross Liftoff Weight, Kg(lb) 1,146,320(2,527,176)	Length, m(ft) 49.1(161.2) Dry Weight, Kg(lb) 103,773(228,777) LH ₂ Weight, Kg(lb) 120,551(265,679) LOX Weight, Kg(lb) 843,579(1,859,755) Gross Liftoff Weight, Kg(lb) 1,117,258(2,463,106)

FIGURE II-1 SSTO VEHICLE SIZES USING DENSIFIED PROPELLANTS

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FIGURE II-2 KSC LC 39 SSTO/DENSIFIED HYDROGEN BASELINE LOADING SYSTEM

the system were evaluated as to its contribution to the overall system enthalpy gain during loading. The sources of enthalpy increase during loading are:

- Storage tank pressurization heating;
- Transfer line chilldown, friction and environmental heating;
- Vehicle tank chilldown, environmental and pressurization heating.

Table II-1. LH₂ Loading Sequence for Shuttle

Operation	Time (Min.)	% Vehicle Load	Transfer Rate m ³ /min (GPM)
Facility and Vehicle Chilldown	10	0	0-3.861(0-1020)
Storage Tank Pressurization and Initial Fill	19	0-2	.382-5.791 (101-1530)
Fast Fill	32	2-98	38.611-45.576 (10,200-12,040)
Topping	12	98-100	3.835-4.221 (1013-1115)
Replenish	45	100	.386-1.16 (102-306)

The heat sources were analyzed parametrically and then combined for a final iterative solution as shown graphically in Figure II-3.

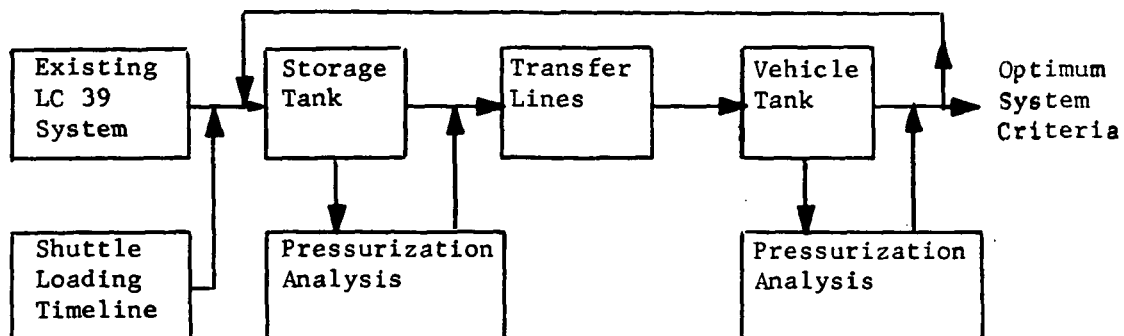


Figure II-3. System Optimization Iteration Process

The loading milestones to be used in this analysis are shown below compared to the present Shuttle LH₂ loading milestones.

Vehicle Loading Milestones

<u>SSTO/SLH₂</u>	<u>SHUTTLE/LH₂</u>
1. Line Chillydown and Storage Tank Pressurization (TBD)	Line Chillydown (10 min.)
2. Initial Fill to 2% of Flight Load (5 min.)	Pressurization and Initial Fill to 2% (19 min.)
3. Fast Fill 100% of Flight Load (TBD)	Fast Fill to 98% (32 min.)
4. Upgrading to 50% SLH ₂ (TBD)	Topping to 100% (12 min.)
5. Pad Hold Maintaining 50% SLH ₂ (45 min.)	Replenish at 100% (45 min.)

The heat load into the normal boiling point liquid hydrogen (NBP LH₂) in the vehicle tank during Shuttle loading results in the boiling of liquid, whereas for the SSTO tank with subcooled SLH₂, it results in the melting of solid hydrogen causing a decrease in the average density. Therefore, the term "topping" has been changed to "upgrading" for SSTO loading and implies increasing the bulk density to that required for liftoff.

For Shuttle loading, the term "replenish" implies the replacement of the boiloff vapor with LH₂. Since the SSTO propellant is subcooled, no boil-off occurs. To maintain steady-state conditions once loaded to 50% SLH₂, less dense propellant must be replaced with that of a higher density. Therefore, for SSTO the term "pad hold" replaces "replenish".

In order to reduce the variables of this analysis, the initial fill sequence was arbitrarily assumed to be 5 minutes and has been separated from the storage tank pressurization. Also, the vehicle tank is filled to 100% instead of 98% at the fast fill rate and this rate is continued for the upgrading sequence. By upgrading at the fast fill rate, the minimum time required to achieve the liftoff density with a minimum of propellant loss is determined. In practice, however, the transition from fast fill to upgrading to pad hold flowrates would be gradual over a finite time interval depending on system design and time constraints.

The analysis also assumes that the storage tank is filled with 60% SLH₂ before the loading sequence begins. This quality was chosen as a result of discussions with the Cryogenic personnel at the National Bureau of Standards, Boulder, Colorado, who indicated this was the highest quality which can be readily produced, stored and transferred. It should be pointed out the vehicle tank could be loaded to 50% SLH₂ with 50% SLH₂ in the storage tank since the solid and liquid can be separated, but the flowrates required would be higher.

1. Storage Tank

The LC 39 liquid hydrogen tank is a vacuum jacketed perlite-insulated double-walled sphere capable of storing 3217.6 m³ (850,000 gal) [227,703 Kg (502,000 lb)] of liquid hydrogen with a 10% ullage for a total volume of 3539 m³ (935,000 gal) (Reference 6). To maintain the same weight, the maximum capacity of 60% SLH₂ is 2756 m³ (728,000 gal) due to its higher density [81.7 versus 70.8 Kg/m³ (5.1 versus 4.42 lb/ft³)]. The inner sphere is fabricated from austenetic stainless steel with an inside diameter of 18.75 m (61.5-ft) at ambient temperature. The outer sphere is carbon steel of 21.3 m (70-ft) diameter. The annular space is maintained at a pressure of 6.67 KPa (50 torr) or less, and the design working pressure of the inner tank is 620.5 KPa (90 psia) with the rated vacuum in the annulus. To determine if this configuration tank is suitable for use in a slush hydrogen system, the following topics were studied and are discussed in the following paragraphs:

a. Storage Tank Heat Leak - The specification boiloff rate for the LC39 LH₂ storage tank is 0.075%/day (maximum) or 2.41 m³/day (637 gal/day)² if fully loaded to 3217.6 m³ (850,000 gal) (Reference 6). From Reference 8 and conversations with NASA/KSC launch operations personnel, it was determined that during Apollo and Skylab missions the actual boiloff rates were approximately 0.76 m³/day (200 gal/day) for the Pad A tank and 3.40 m³/day (900 gal/day) for the Pad B tank.

By multiplying these values by the LH₂ density and heat of vaporization (Reference 9), and dividing by the internal sphere area [1103.7 m² (11,880 ft²)] the corresponding heat leak rates are calculated. These values are 0.801, 0.252 and 1.13 w/m² (0.254, 0.080 and 0.359 Btu/ft²/hr) for specification, Pad A, and Pad B, respectively.

If the tank were filled with 60% SLH₂ and it is assumed that all heating goes into melting solid hydrogen, then the quality decay as a function of time can be determined as shown in Figure II-4. The quality decays to 50% in 58 days in the Pad A tank and in 13 days

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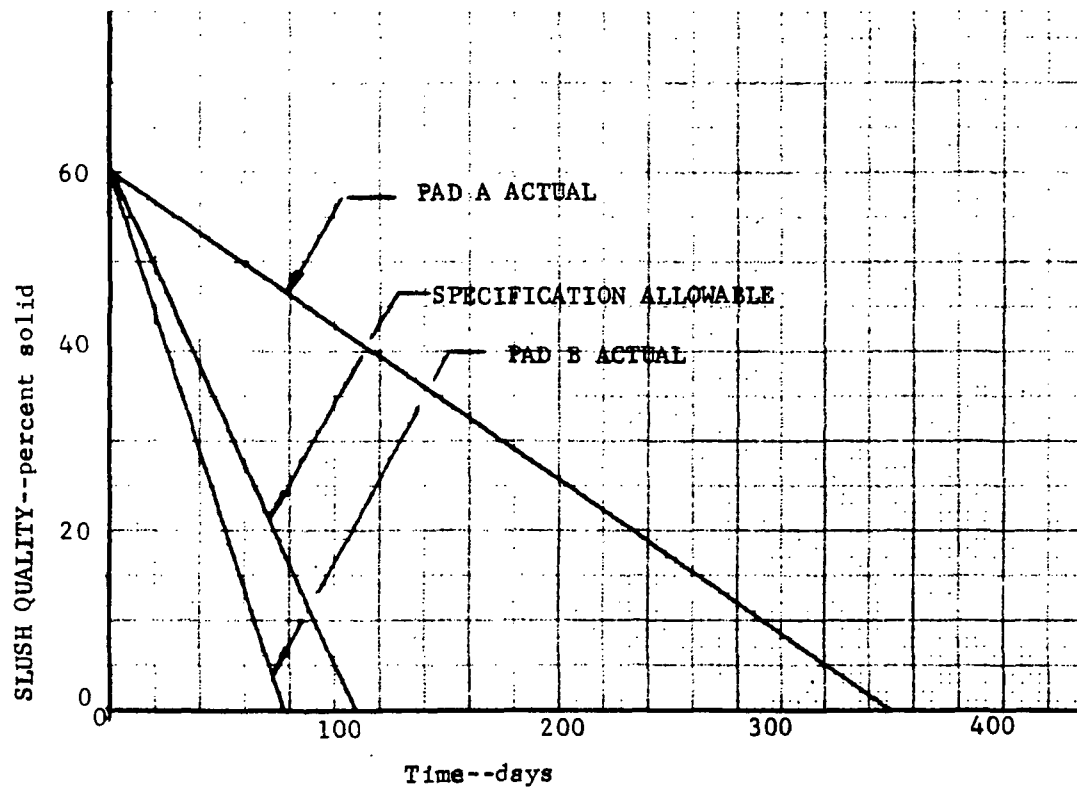


FIGURE II-4 SLUSH HYDROGEN QUALITY DECAY IN THE
LC 39 LH₂ STORAGE TANKS

in the Pad B Tank.

Within the past year, these tanks have been refinished with a darker rust preventative compound that has resulted in a higher boiloff rate. Since the emphasis of this study will be to minimize heat leaks and the above values are achievable, they will be used. Also, since the Pad A tank heat leak is lower and, therefore, more desirable it will be used for any subsequent system heating analysis in this report. In view of the anticipated minimum SSTO launch rate (Reference 1) of one launch every 15 days, the Pad A tank heat leak rate is acceptable.

b. Storage Tank Standby Pressure - During normal standby conditions with NBP LH_2 the storage tank vents to the atmosphere through a check valve that maintains the sillage pressure at 1.05 KPa (0.5 psig). With SLH_2 and its vapor pressure at triple point [7.03 KPa (1.02 psia)], the tank would be subjected to a negative delta pressure under equilibrium conditions. This will cause no problem, however, since the tank was designed to withstand a full vacuum in the inner sphere (Reference 6) and was initially filled by evacuating and backfilling with LH_2 .

Transfer of SLH_2 into the storage tank during its filling or out of the tank during vehicle loading requires that the system be pressurized above atmospheric pressure. Accordingly, with operation above atmosphere pressure required, there is no need to maintain SLH_2 at triple point pressure.

Therefore, it is recommended that the SLH_2 storage tank be designed to allow for triple point pressure storage, but the tank pressure would normally be at or above one atmosphere.

c. Storage Tank Pressurization and Stratification - Most liquid hydrogen systems transfer by means of gaseous hydrogen pressurized expulsion due to its low density and relative simplicity of the system. This includes the existing LH_2 system at LC 39 as well as the roadable tankers used for transporting the fuel from the production plant. A major question at the outset of this study, therefore, was the feasibility of transferring subcooled hydrogen by GH_2 pressurized expulsion since its vapor pressure is less than one atmosphere.

To analyze this problem, the Martin Marietta Cryogenic Tank Pressurization/Stratification computer program entitled PRESS was utilized. The program was modified to account for the solid hydrogen and retitled SLUSHPRESS. The model assumes the tank is initially filled with a homogeneous mixture of SLH_2 . The heat from pressurization forms a stratified layer of warmer liquid on top of the bulk slush, and the

bulk slush remains at the original consistency (60% solid fraction for the baseline) thereby allowing for the outflow of high quality propellant from the bottom of the tank. This latter phenomenon has been verified experimentally as reported in Reference 10. The program computes, the stratified layer weight and temperature, the ullage gas pressure and temperature and the weight of total propellant remaining as functions of heat and mass transfer effects. A detailed description of the program is included in Appendix A.

An initial series of runs were made to determine the pressurant gas flowrate required for initial tank pressurization and to sustain flow. The existing LC 39 LH₂ tank pressurization system (vaporizer) is composed of a 10 cm (4-inch) diameter, schedule 40 aluminum coil 793 m (2600-ft) long (Reference 8). Liquid drawn from the bottom of the tank is fed into this coil by valve control. The gas effluent from the coil is fed into the tank ullage for pressurization of the tank. The driving force for the coil is the liquid head in the storage tank. The coil was designed for a maximum flow capacity of 816 Kg/min (180 lb/min) at an operating pressure of 618 KPa (75 psig) and supplies gas to the storage tank at 72.2°K (130°R). The SLUSHPRESS program was input with data defining the existing storage tank configuration and capacity loaded with 60% SLH₂. The tank outflow profile was input corresponding to the Shuttle loading flowrates. The GH₂ pressurant gas temperature was assumed to be the same as the current system [(72.2°K (130°R))]. The storage tank pressure was assumed to be 618 KPa (75 psig) [620 KPa (90 psia)] since the storage tank pressure required to transfer SLH₂ to the SSTO was not known at this point.

Three runs were made at maximum pressurant gas flowrates of 81.6, 136 and 191 Kg/min (180, 300 and 420 lb/min) with the results shown in Figure II-5. In order to initially pressurize to 620 KPa (90 psia) in approximately 5 minutes and to sustain 620 KPa (90 psia) during outflow, a pressurant gas flowrate on the order of 181 Kg/min (400 lb/min) or double the existing system capacity is required. The pressurant gas flowrate will be re-evaluated later as the actual pressure and flowrates for the SLH₂ system are defined. These initial runs, however, did demonstrate that pressurized expulsion of slush hydrogen with gaseous hydrogen pressurant was feasible.

The next major concern was the effect that this warm pressurant gas had on the bulk slush and how much solid hydrogen was melted. To investigate this matter, the program was modified to tabulate the heat transferred from the liquid film (stratified layer) into the slush. The same tank data, loading profile and operating tank pressure [620 KPa (90 psia)] were input to the program. The results, when plotted (Figure II-6) and integrated, revealed that more solid

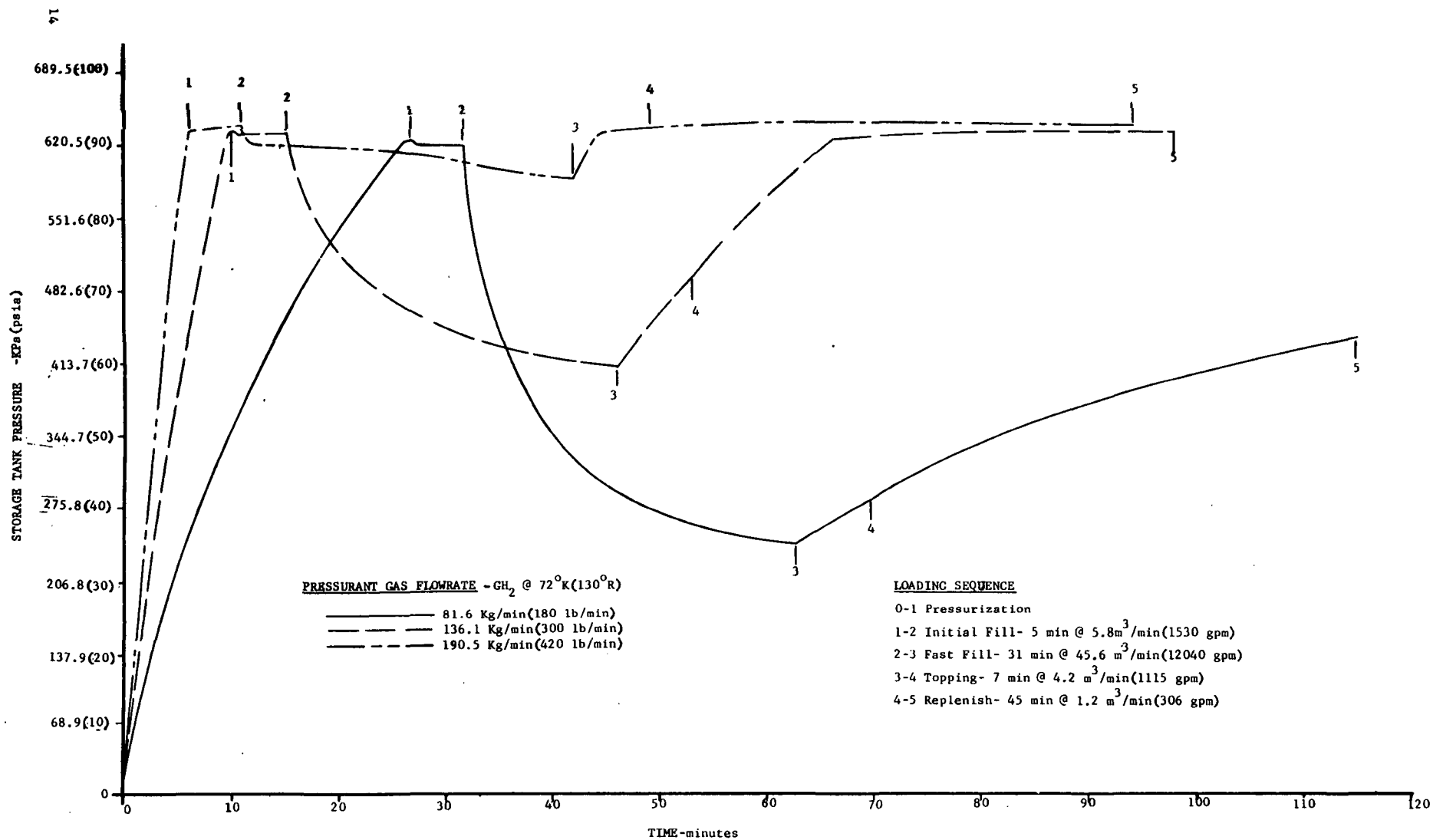


FIGURE II-5 SLH₂ STORAGE PRESSURIZATION FOR VARYING PRESSURANT FLOWRATES

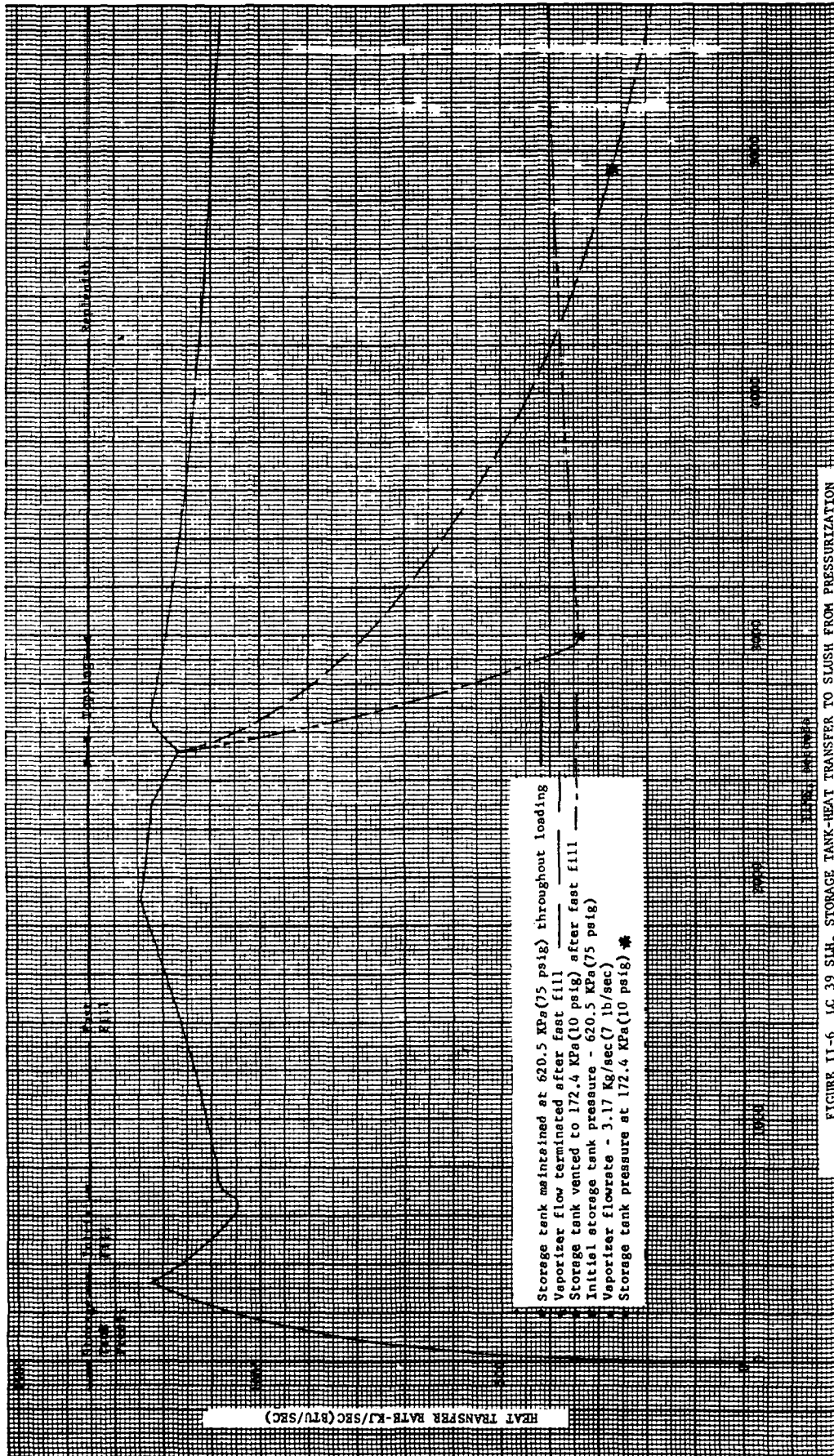


FIGURE II-6 LC 39 SLH₂ STORAGE TANK-HEAT TRANSFER TO SLUSH FROM PRESSURIZATION

would be melted $[1334.68 \text{ m}^3 (352,586 \text{ gal})]$ than would be left in the tank $[602.058 \text{ m}^3 (159,047 \text{ gal})]$ after the SSTO was loaded. The total heat transferred into the bulk slush $[6.71 \times 10^6 \text{ KJ} (6.36 \times 10^6 \text{ Btu})]$ divided by the solid hydrogen heat of fusion $[58.1 \text{ KJ/Kg} (\text{Btu/lb})]$ equals 115466 Kg (254,560 lb) or $1334.68 \text{ m}^3 (352,586 \text{ gal})$ of solid hydrogen.

In order to reduce this heat transfer it was decided to reduce the storage tank pressure to 172 KPa (25 psia) after fast fill allowing since it requires significantly less pressure to maintain flowrates for the topping and replenish modes of Shuttle. The program input was changed to terminate the vaporizer flow after fast fill allowing the tank ullage pressure to decay during topping and replenish flow. The storage tank pressure decayed below 170 KPa (10 psig) prior to the nominal end of replenish indicating the vaporizer supply valve must be reopened to maintain 170 KPa (10 psig). From this plot, it was calculated that $4.61 \times 10^6 \text{ KJ} (4.37 \times 10^6 \text{ Btu})$ were transferred into the bulk slush resulting in the melting of $916.493 \text{ m}^3 (242,112 \text{ gal})$ of solid hydrogen.

Finally, the program input was changed to vent the tank to 170 KPa (10 psig) at the end of fast fill and then maintain tank pressure constant at 170 KPa (10 psig). This case would reduce the heat transfer more rapidly, would provide a more constant upstream pressure for regulating the replenish valve flowrate (see Figure II-2), but would waste the vented hydrogen gas. From this plot, it was calculated that $4.38 \times 10^6 \text{ KJ} (4.15 \times 10^6 \text{ Btu})$ were transferred into the bulk slush resulting in the melting of $870.353 \text{ m}^3 (229,923 \text{ gal})$ of solid hydrogen.

In order to allow for the melting of $870.645 \text{ m}^3 (230,000 \text{ gal})$ of solid hydrogen and still deliver 50% slush at the end of vehicle loading, a storage tank of $4542.5 \text{ m}^3 (1,200,000 \text{ gal})$ capacity initially at 60% quality would be required.

The preceding discussions of pressurant gas flowrates and heating of the bulk slush apply only for a system operating pressure of 620 KPa (90 psia) and the Shuttle loading rates. They have been included in this report to show how this study developed and should not be considered as final results.

To develop the parametric relationship between storage tank pressure and the heat transferred to the propellant in the tank during loading, several computer runs were made with the pressure and flowrate data in Table II-2. The storage tank pressure required for transfer is the summation of the transfer line pressure drop and SLH_2 elevation head

between the storage tank and the fully loaded SSTO tank. The transfer line pressure drop for varying flowrates and line sizes is discussed later (Para. I.A.2.a). The elevation head, shown in Figure II-2 is 73.8 cm (242-ft) of SLH_2 or approximately 59 KPa (8.6 psi).

Table II-2. SLH_2 Storage Tank Heat Input

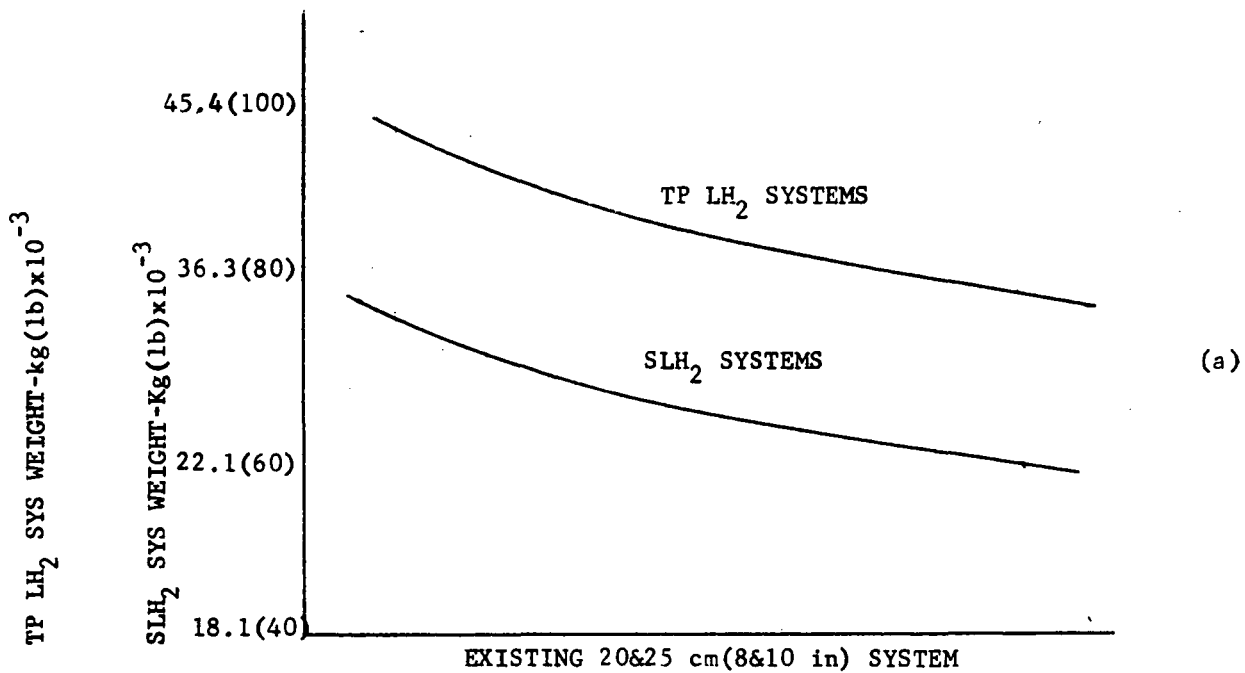
Transfer Line Size (Inch) cm	Fast Fill Flowrate m^3/min (gpm)	Storage Tank Ullage Press. KPa (psia)	Heat Input $KJ \times 10^6$ ($Btu \times 10^6$)
20/25 (8/10)	37.9 (10,000)	448 (65)	3.9 (3.7)
25 (10)	39.8 (10,500)	414 (60)	3.4 (3.2)
30.5 (12)	41.6 (11,000)	290 (42)	2.8 (2.7)
35.6 (14)	45.4 (12,000)	269 (39)	2.5 (2.4)

The results are plotted in Figure II-7 and show the weight and temperature of the stratified layer at the end of the SSTO loading sequence as a function of transfer line size.

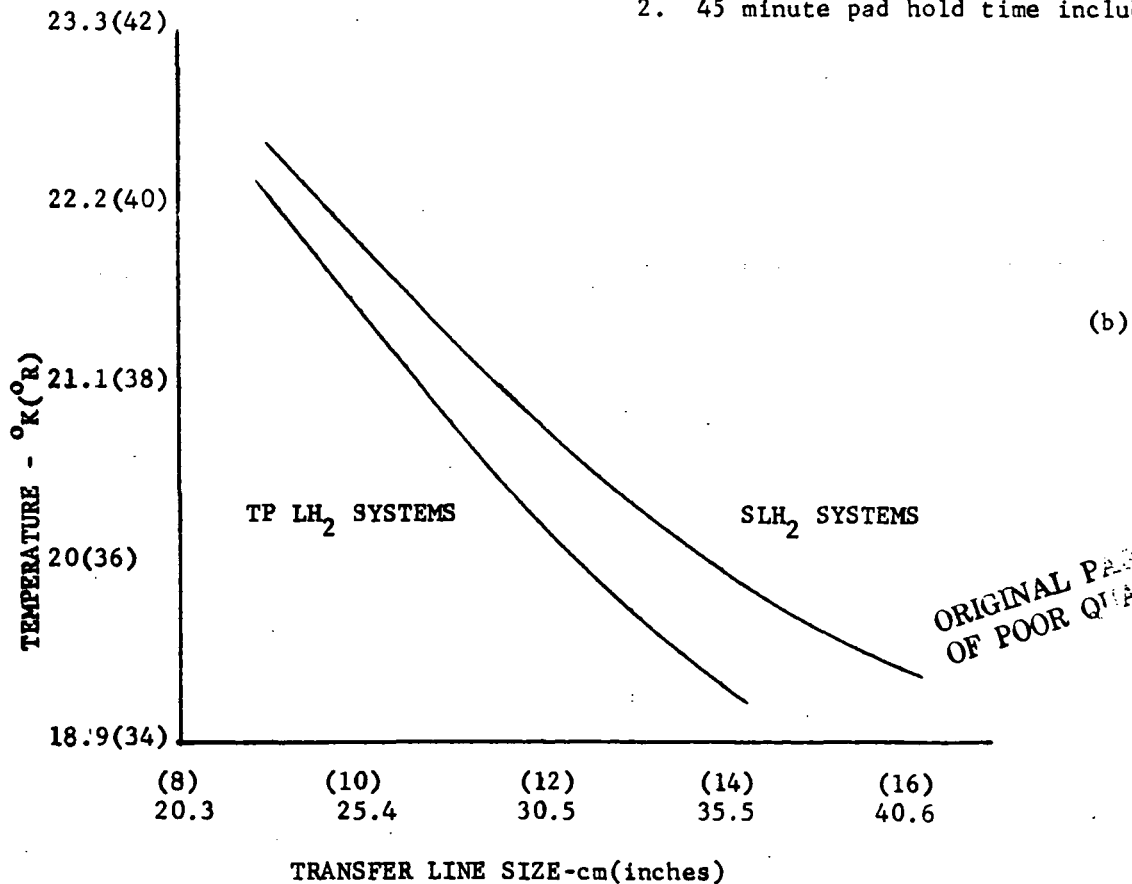
Since the tank is full of 60% SLH_2 initially, the heat input from pressurization can be calculated by subtracting the final enthalpy of the stratified layer from the enthalpy of 60% SLH_2 [3.46 KJ/Kg (148.8 Btu/lb)] and multiplying by the stratified layer weight.

d. Slush Mixing in Storage Tank - Mixing of slush in hydrogen tanks has been accomplished in small containers (one cubic meter) with propellers located inside the tank and by discharging high velocity slush into the tank through a duct (Reference 11). Mixing of SLH_2 with propellers in tanks the size of those at LC 39 does not appear feasible for the following reasons:

- Large size of tank and quantity of fluid would require long shafts, large propellers and bearings.
- Inaccessibility to inside of tank or annular space for propeller, bearing and motor maintenance.
- Heat leak associated with passing propeller drive shaft through tank walls.
- A system for circulating SLH_2 will be required for storage tank quality upgrading and could be used for mixing as well.



- NOTES: 1. STORAGE TANK INITIALLY FILLED TO 2756 m³ (728,000 gal) 60% SLH₂ OR 4414 m³ (1,166,000 gal) TP LH₂.
2. 45 minute pad hold time included.



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FIGURE II-7 STORAGE TANK STRATIFIED LAYER WEIGHT AND TEMPERATURE AT END OF SSTO LOADING VS LINE SIZE

Therefore, a system which withdraws triple point liquid from the tank and either flows through an upgrading processor or directly back into the tank was considered.

Experimental data with small containers have indicated that a mixing rate of 1/12 of the fluid volume per second is required for ducted mixing of 55% SLH₂ (Reference 11). For a SLH₂ volume of 2755.8 m³ (728,000 gal),² the required mixing flowrate would be 229.6 m³/second (60,666 gal/second) or over 1.1 x 10⁴ m³/min (3 x 10⁶ gpm). This rate is obviously excessive and is not considered realistic since data used for this analysis was determined subjectively in a small (1 m³) container.

In order to gain better understanding of the mixing problem and slush H₂ in general, a meeting was held at the National Bureau of Standards, Cryogenics Division, Boulder, Colorado with Messrs. C. Sindt and R. Voth. Through the discussion and the viewing of films of the flow, settling and mixing of SLH₂, the following was learned:

- Flow of slush with mass fraction as high as 0.7 has been observed.
- When mixing in a container is stopped, solid particles settle to the bottom very rapidly.
- To reinitiate flow in containers of settled, aged slush, even with as high as 0.7 mass fraction, very little, if any, agitation is required.
- Aged slush characteristics seem to remain constant after approximately 30 hours in a well-insulated container and approximately 2 hours in a poorly-insulated container.
- A device has been developed and is being tested which determines the average density of a fluid in a container by measuring the resonant frequency of the tank from a single point within the tank.

If this average density measuring device proves functional and little or no agitation is needed to initiate flow, then there will be no need to maintain a homogenous mixture of SLH₂. There is probably a need for a system to mix the bottom of the storage tank at the discharge duct but the flowrate required should be low.

It is envisioned that this system would drain triple point liquid from the bottom of the tank through a high speed pump located outside the tank. The pump discharge would be returned to the tank

and ducted across (normal to) the transfer line outlet so as to cause mixing. To conserve solid content within the tank, a 30-mesh screen would be positioned over the pump suction line. This allows the separation of liquid and solid as the liquid will pass through the screen and solid will not. This phenomena has been experimentally verified and reported in Reference 12.

The use of a 30-mesh screen would also apply to the vaporizer outlet duct so that only triple point liquid was consumed for pressurization.

Experimental testing with dewars significantly larger than those from which current test data has been derived must be conducted to validate these observations and to determine the flowrates required to insure no clogging of discharge ducts.

2. Transfer Line

The information in the previous section showed that a storage tank of similar design to the existing LC 39 tank can deliver high quality (60%) SLH₂ at the tank outlet. The next item to consider is the transfer line between the storage and vehicle tanks and to establish the line configuration that will minimize heat input (enthalpy gain) to the propellant during loading. The sources of propellant heating in the transfer line are friction heating from line and components, environmental heating through the line insulation and initial line chilldown.

The existing LC 39 transfer line is a vacuum jacketed, multilayer insulated (MLI) pipe whose length from the storage tank to the vehicle tank is approximately 535 m (1750 ft). The majority of the line is 25.4 cm (10-inch) nominal diameter with approximately 15 m (50 ft) of 20 cm (8-inch) pipe in the vehicle and on the mobile launcher platform (MLP).

The inner line is made of Schedule 5 Invar pipe and the insulation consists of 20 layers of aluminized mylar film and dacron mesh spacer wrapped around the inner pipe (Reference 6). This is the baseline configuration for the analyses that follow.

- a. Transfer Line Pressure Drop - The heat input and corresponding quality decay in the transfer line due to friction is a function of total system pressure drop. For the baseline system shown in Figure II-2 the total line and component pressure drop was calculated for varying flowrates and line diameters and is presented in Figure II-8.

- NOTES: 1. TOTAL GSE AND VEHICLE PIPING AND COMPONENTS PRESSURE DROP
2. LINE LENGTH = 533.4 m (1750 ft)

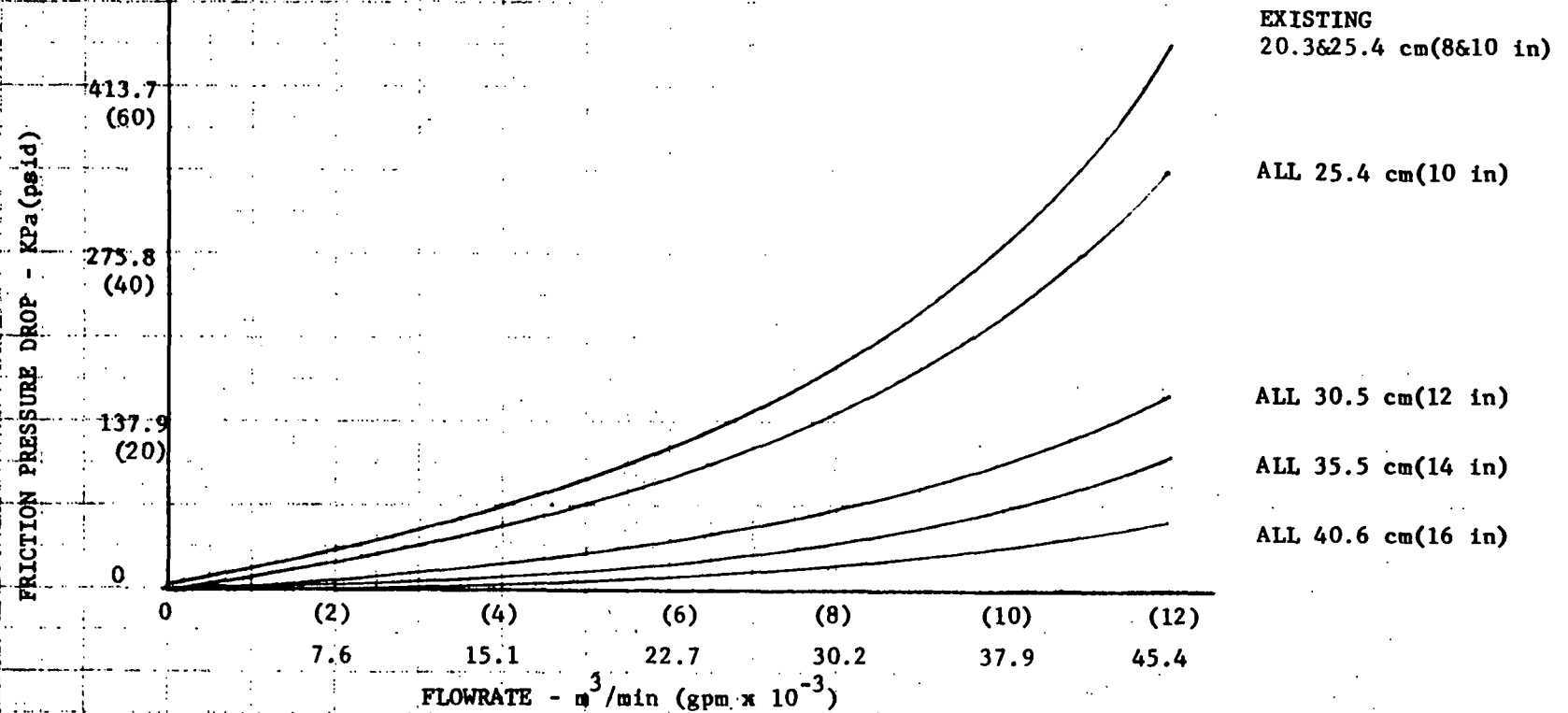


FIGURE II-8 SSTO SLH₂ & TP LH₂ TRANSFER SYSTEMS-FLOWRATE VS PRESSURE DROP

For the ground system (from storage tank outlet valve A3301 to the vehicle interface) the pressure drop attributable to line friction loss was calculated using the standard Darcy equation (Reference 13) and the system L/D values defined in Reference 14. The friction factor was calculated using data presented in Reference 15 extrapolated to a Reynolds number of 1×10^7 . The pressure drop for system valves was calculated by using the C_v values in Reference 6. These values are for NBP LH_2 but are applicable to slush hydrogen as it has been demonstrated that pressure losses in systems flowing slush hydrogen of solid fractions to 0.5 are essentially the same as with liquid (Reference 15). Due to the uncertainty of the actual SSTO fill line configuration the vehicle line pressure drop used was the known Shuttle ICD pressure drop requirement (Reference 7) converted to slush density.

The existing LC 39 LH_2 system has a filter in the transfer line between the fill valve and the vehicle interface not shown in Figure II-2. Since the filter element is designed to remove all particles larger than 70 microns in diameter (Reference 6) the solid hydrogen particles would be restricted. Therefore, slush hydrogen cannot be filtered between the storage and vehicle tanks by conventional means and no filter pressure drop is included. The matter of slush filtration is of major concern, however, and will be discussed later (Para. II.E.2).

- b. Transfer Line Thermal Analysis - A thermal analysis was performed to determine the optimum insulation system for a slush hydrogen transfer line. Also, published data of heat leak rates for the existing LC 39 lines were conflicting and the analysis provided an average heat leak rate value that could be used with confidence in this study. The analysis considered three different insulation configurations, namely: vacuum jacketed, vacuum jacketed with multilayer insulation (MLI) and active cooling. The heat transfer effects of system components, joints, and fittings were also included.

The detailed analysis is included as Appendix B of this report. The results show that the vacuum jacketed line with MLI, is the best choice for slush hydrogen. Also, the average environmental heat leak rate for SLH_2 transfer is 0.38 w/m of line/cm line diameter (1 Btu/hr/ft of line/inch line diameter) or 9.6 w/m (10 Btu/hr/ft) for the existing 25 cm (10-inch) cross-country transfer line.

- c. Transfer Line Heating (Friction, Environmental and Chillover)- With the transfer line pressure drop and heat leak defined, the

loss in SLH₂ quality during transfer resulting from friction and enviromental heating can be calculated. This applies to steady state transfer conditions. The line chilldown energy will be considered separately.

Assuming that all heat goes into melting solid hydrogen, the loss in quality due to friction can be calculated from equation (1).

$$\Delta X_{\text{friction}} = \left(\frac{144 \Delta P}{778 \rho L_f} \right) \quad (1)$$

where: ΔX = change in solid content (%)
 ΔP = friction pressure drop (psid)
 ρ = slush density (lb/ft³)
 L_f = latent heat of fusion (Btu/lb)

Again, assuming that all heat goes into melting solid hydrogen, the loss in quality due to enviromental heating can be calculated from equation (2).

$$\Delta X_{\text{env.}} = \left(\frac{3600 q L}{\dot{V} L_f} \cdot \frac{\dot{q} L}{\dot{V} L_f} \right) \quad (2)$$

where: q = transfer line heat leak (Btu/hr-ft)
 L = length of line (ft)
 \dot{V} = volumetric flowrate (ft³/hr)

The summation of these two expressions is the change in quality between the storage and vehicle tanks and is plotted in Figure II-9 as a function of flowrate for varying line sizes. It is interesting to note that environemntal heating is the major contributor to quality loss at the lower flowrates as represented by the negative sloping lines. Friction heating is the major source of heat a higher flowrates as represented by the positive sloping lines.

The net steady-state heat transferred into the propellant for a particular size system at a given flowrate is determined by equation (3)

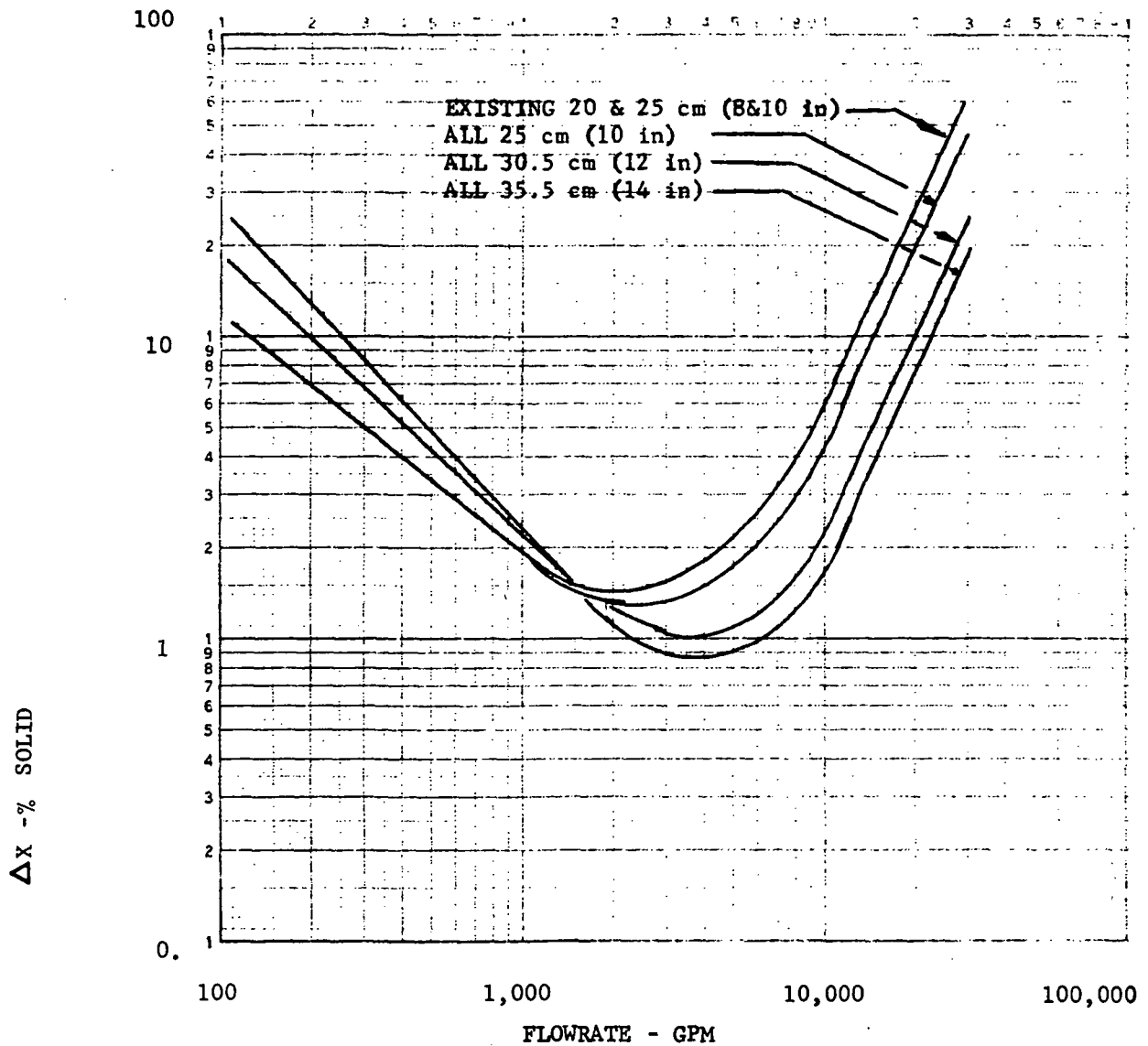
$$Q = X V \rho L_f t \quad (3)$$

where: Q = net heat input (Btu)
 t = time (hr)

The heat transferred to the propellant for line chilldown is a function of the mass and heat capacity of the inner line and components, the insulation efficiency, the outer jacket support configuration and the transfer duration. Published data on chilldown of MLI vacuum jacketed lines (Reference 16) shows that over 95% of the chilldown energy has been removed from the line after 100 minutes and steady-state heat transfer conditions essentially exist. With a minimum 45-minute pad hold requirement the time for SSTO SLH₂ loading will be approximately 100 minutes. An analysis of the transient chilldown of the existing 20/25 cm (8/10-inch) transfer line at LC 39 has been studied by computer model and reported in Reference 14. This analysis determined that the total chilldown energy is 1.7×10^6 Btu for the existing system. For systems of line size other than the existing 20/25 cm (8/10-inch) the chilldown energy is approximated by multiplying this value by the ratio inner line weight per unit volume.

- d. Transfer Line Size - In Figure II-9 it is noted that for flowrates less than $5.68 \text{ m}^3/\text{min}$ (1500 gpm) is a decrease in pipe diameter reduces the quality loss. This is due to the decrease in surface area of the smaller pipe which reduces environmental heat transfer. At flowrates above $5.68 \text{ m}^3/\text{min}$ (1500 gpm), an increase in pipe diameter reduces quality loss because the greater cross-section area reduces pressure drop. It will be shown later in this report that flowrates in excess of $5.68 \text{ m}^3/\text{min}$ (1500 gpm) are required to load, upgrade and maintain quality during pad hold for SSTO loading. Therefore, to minimize quality loss during transfer, the transfer line size should be increased.

Also, in Figure II-9, it is noted that for a given flowrate a disproportionate reduction in quality loss occurs for a line size increase from 25 to 30.5 cm (10 to 12 inches). This is simply a result of the relative increase in cross-section area of standard pipe. Line sizes of 35.5 cm (14 inches) and larger would require outer jackets 40.5 cm (16-inch) and larger and practical considerations such as fabrication, installation and cost offset the advantage. Therefore, since an increase to a 30.5 cm (12-inch) line shows a significant advantage and is considered within practical limits it, along with existing 20 and 25 cm (8 and 10 inch)[20/25 cm (8/10 inch)] line, will be used for further system definition.



- NOTES: 1. TRANSFER SYSTEM LENGTH = 533.4m (1750 ft)
2. 25 cm (10 in) PIPE HEAT LEAK = 9.62w/m (10 BTU/HR-FT)
3. VEHICLE PIPING HEAT LEAK INCLUDED

FIGURE II-9 SLH₂ QUALITY DEGRADATION IN GROUND TRANSFER SYSTEM-
TOTAL FRICTION AND ENVIRONMENTAL HEATING FOR VARYING
SIZE TRANSFER LINES

- e. Fluidizing Velocity in Transfer Line - A question posed at the outset of this study was - could slush be transferred in pipes the size required for SSTO loading or would the solid particles settle in the bottom of the pipe and triple point liquid flow above it? The fluidizing velocity or critical flow rate of 50% SLH₂ has been determined experimentally to be .46 m/sec (1.5 ft/sec) (Reference 15). This equates to a volumetric flow-rate of 1.53 m³/min (403 gpm) in 25 cm (10-inch) pipe and 2.15 m³/min (568 gpm) in 30.5 cm (12-inch) pipe which is well below the SSTO loading rates. These values should be used with caution, however, since the critical velocity was determined from small scale [16mm ID (5/8-inch ID)] testing and may not be accurate for line diameters of this size. Larger scale testing must be conducted to validate this analysis.

3. Vehicle Tank

- a. Insulation - The fuel tank for the baseline SSTO (SLH₂/TP LOX) vehicle as defined by Reference 1, is an integral multi-lobe aluminum structure which conforms to the forward fuselage shape and provides the primary structural load paths of the vehicle. The tank volume is 1519.2 m³ (53,650 ft³) and contains 120509 Kg (265,677 lb) [1474.9 m³ (389,630 gal)] of 50% SLH₂ propellant at liftoff. The heat transfer area of the tank is 876 m² (9429 ft²).

A detailed analysis of the insulation system requirements for this tank are included as Appendix C. Among other parameters the analysis evaluates the use of internal versus external insulation and the economics of capillary versus foam internal insulation. It is recommended that an internal insulation of 2.5 cm (1-inch) thick polyphenylene oxide (PPO) foam be used for the SSTO tank. This equates to an average heat flux of 709 w/m² (225 Btu/ft²/hr). The heat input to the propellant in a PPO foam insulated SSTO tank is shown in Figure II-10 as a function of insulation thickness and total loading time. This value is the total heat input from tank chilldown and environmental heat leak.

For the 1476 m³ (390,000 gal) SSTO fuel tank the total loading time is a function of the fast fill rate, assuming a constant 45-minute pad hold period. The resulting heat input to the propellant of a 2.5 cm (1-inch) PPO foam insulated vehicle tank is tabulated in Table II-3.

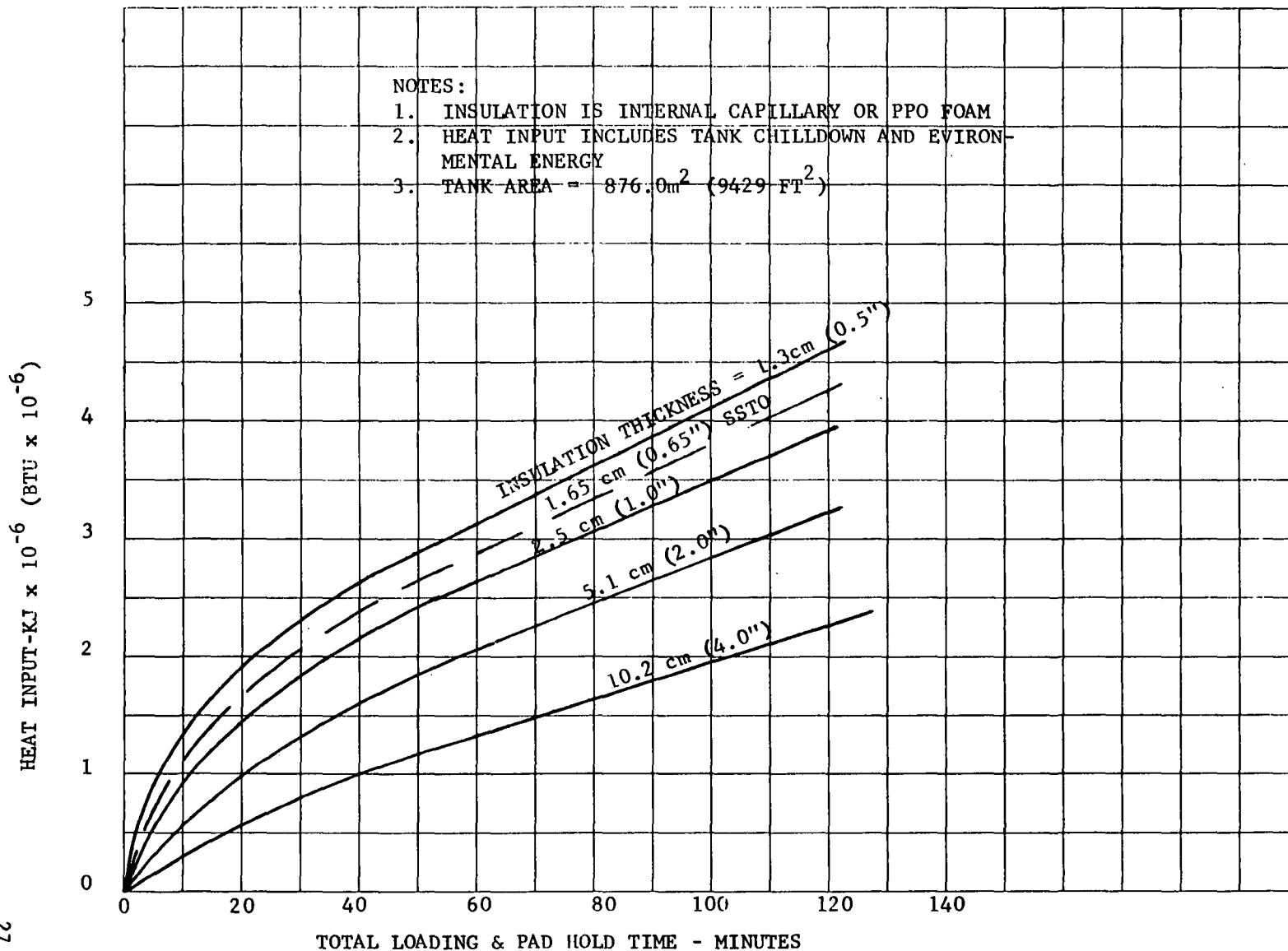


FIGURE II-10 SLH₂ & TP LH₂ SYSTEM HEAT INPUT-SSTO TANK HEAT VS LOADING TIME

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Table II-3. SLH₂ Vehicle Tank Heat Input (1-Inch PPO Foam)

Fast Fill Rate m ³ /min (gpm)		Total Loading Time (Min.)	Heat Input KJ x 10 ⁶ (Btu x 10 ⁶)	
22.7	(6,000)	110	3.90	(3.70)
30.3	(8,000)	94	3.53	(3.35)
37.75	(10,000)	84	3.32	(3.15)
45.42	(12,000)	78	3.17	(3.00)

The insulation analysis in Appendix C selects PPO foam over capillary insulation primarily on the basis of reduced weight. An unknown at this point, however, is the durability of foam insulations in view of the reusable aspect of the SSTO vehicle. Capillary type insulations are inherently more durable and may prove advantageous. Experimental testing should be conducted to evaluate insulation durability.

- b. Pressurization and Stratification - Pressures in the Shuttle ET during loading of NBP LH₂ are above ambient pressure. As a result, the pressure differential across the tank walls is positive and tank implosions cannot occur. Negative pressures can only be caused by transient phenomena such as geysering which can be controlled or eliminated. Tank implosion can occur when loading subcooled SLH₂ in the SSTO if active pressurization is not provided.

To determine the SSTO pressurization requirement, the SLUSHPRESS program (Reference Appendix A) was input with the vehicle tank configuration data. The model was programmed to calculate the pressurant gas flowrate required to maintain the tank pressure at 103 KPa (15 psia). The pressurant gas utilized was gaseous helium at ambient temperature. Also, by varying the incoming propellant conditions (quality, flowrate, time) the resulting effects on the vehicle load were evaluated and the optimum loading timeline was determined.

4. Total SLH₂ System Transfer Analysis

The preceeding sections defined the system components and discussed the relative effect each has on the total system heat input during loading. The heat input from the storage tank results primarily from the storage tank pressurization and can be reduced by decreasing the transfer flowrate which in turn reduces the required storage tank pressure. The heat input from the transfer line results primarily from friction heating and can also be reduced

by reducing the transfer flowrate. The heat input from the vehicle tank results primarily from environmental heat transfer and can be reduced by decreasing the loading time which means increasing the transfer flowrate.

With the system heat sources defined parametrically the optimum loading rates for SLH_2 can be determined and the loading sequence established for varying size transfer lines. The total system enthalpy gain can then be analyzed and the SLH_2 consumption per launch determined.

- a. Fast Fill Transfer Rate - The system heat input from ground and vehicle sources is shown in Figure II-11 as a function of flowrate. The transfer system heat curves are the total of the storage tank pressurization and transfer line friction and environmental heating for varying transfer line sizes. The vehicle tank heat is plotted for varying thicknesses of internal PPO foam. For a 2.5 cm (1-inch) PPO foam insulated tank, the total system heating is shown in Figure II-12, for varying transfer line sizes as a function of flowrate.

The optimum fast fill flowrate for a particular size system is that which results in the minimum heating. Therefore, for the existing 20/25 cm (8/10 inch) system the optimum fast fill rate for transferring 60% SLH_2 to the SSTO vehicle is 37.85 m³/min (10,000 gpm) and for a 30.5 cm (12-inch) system the fast fill rate is 41.6 m³/min (11,000 gpm).

As mentioned previously, in order to reduce the variables of this analysis and to determine the maximum time required to achieve 50% SLH_2 for liftoff, the fast fill rate is sustained for the upgrading sequence.

- b. Pad Hold Transfer Rate - The flowrate of propellant required to maintain 50% SLH_2 during steady-state pad hold conditions is a function of the vehicle tank heat flux and the delivered propellant quality and is plotted in Figure II-13. For an average heat flux of 709 w/m² (225 Btu/ft²/hr) 2.5 cm (1-inch PPO foam) the 20/25 cm (8/10 inch) system pad hold flowrates are 11.36 m³/min (3000 gpm) for the 20/25 cm (8/10-inch) system at a delivered quality of 58.3% (see Figure II-9) and 11.32 m³/min (2990 gpm) for the 30.5 cm (12-inch) system at a delivered quality of 58.9%.
- c. SSTO/50% SLH_2 Loading Sequence - The loading sequence for the 50% SLH_2 -fueled SSTO vehicle was determined by utilizing the tank pressurization/stratification model for both the storage tank

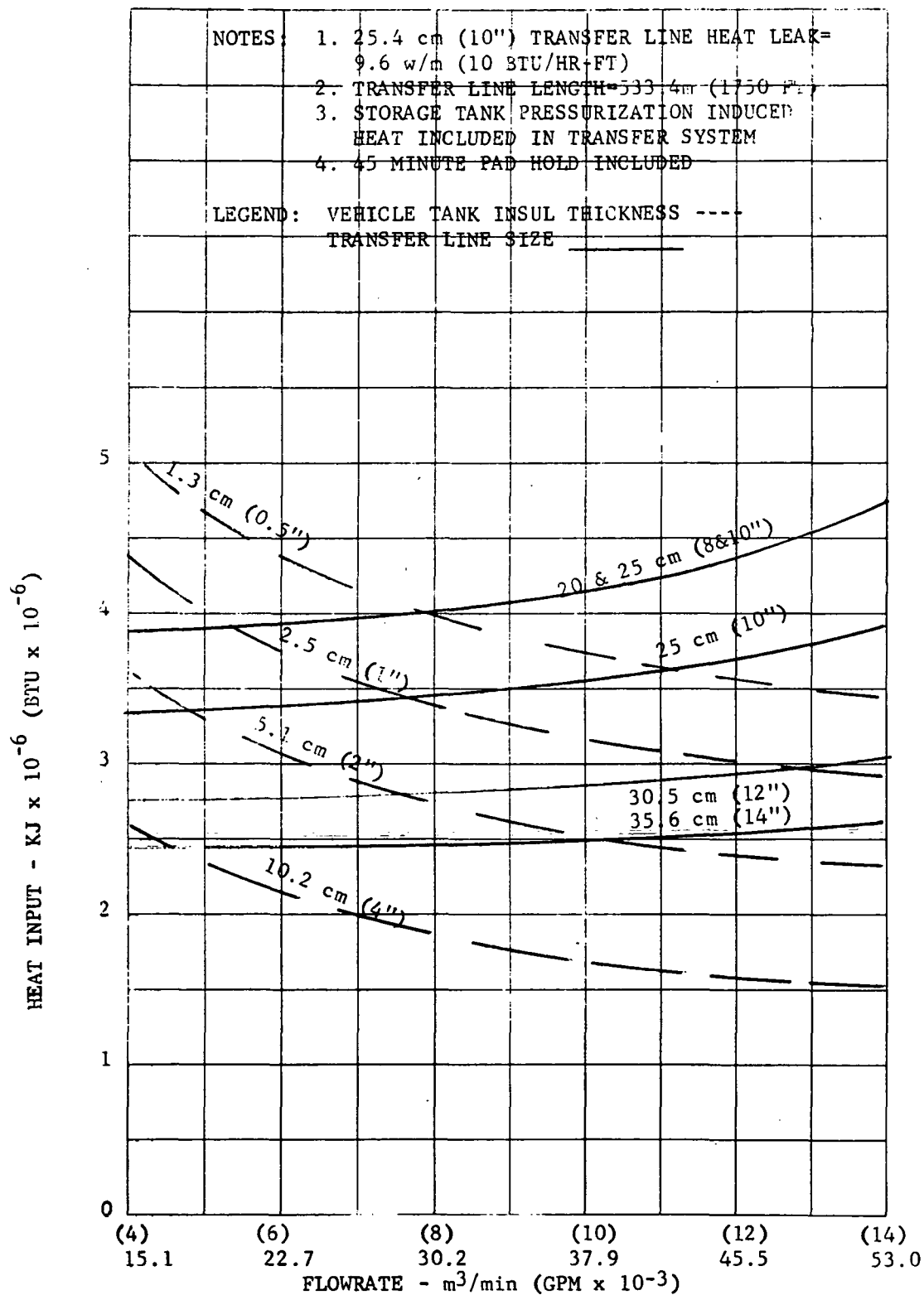


FIGURE II-11 SSTO/SLH₂ SYSTEM HEAT INPUT-
 VEHICLE TANK AND TRANSFER SYSTEM
 HEAT VS FAST FILL FLOWRATE

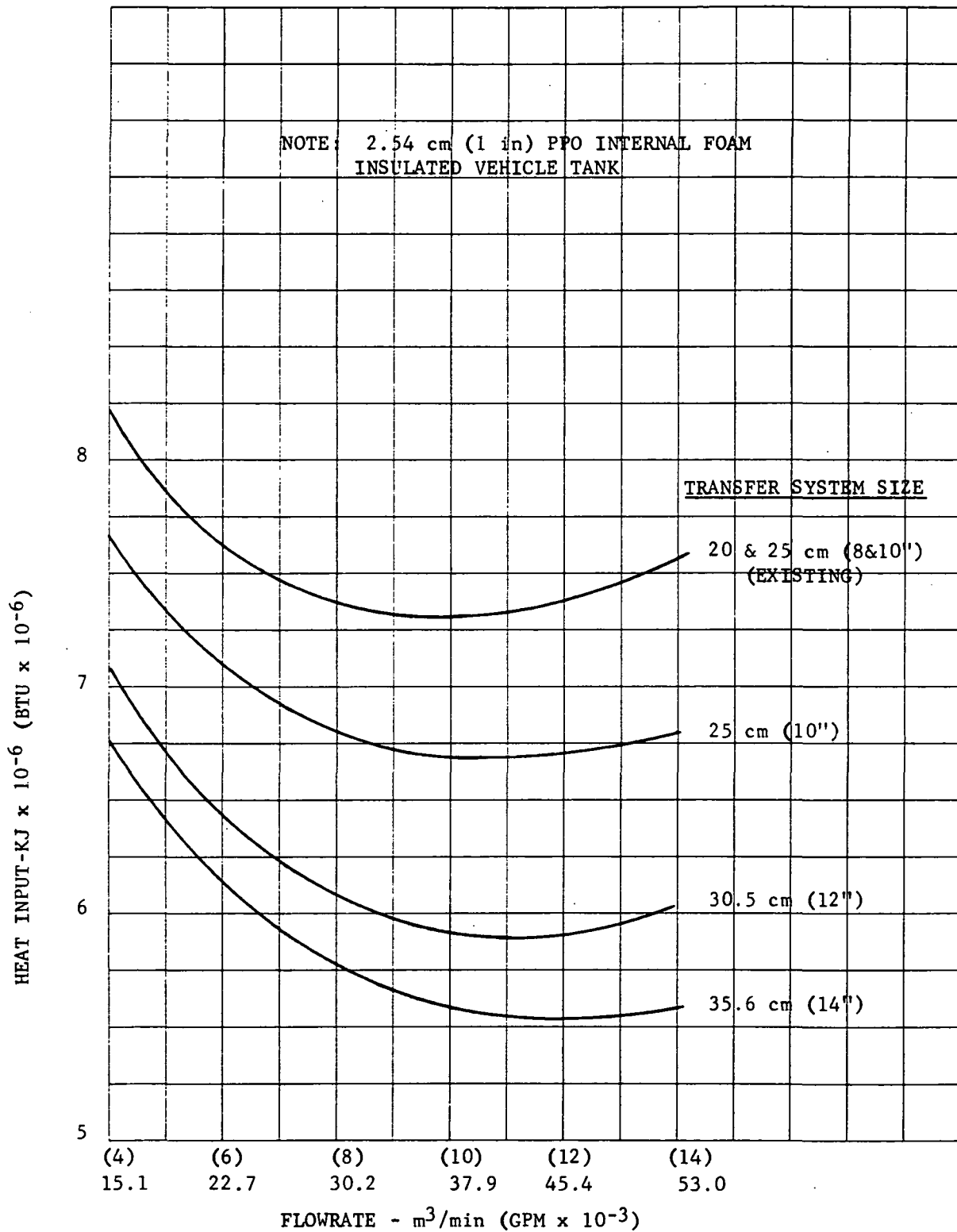


FIGURE II-12 SSTO/SLH₂ SYSTEM HEAT INPUT - TOTAL
HEAT VS FAST FILL FLOWRATE

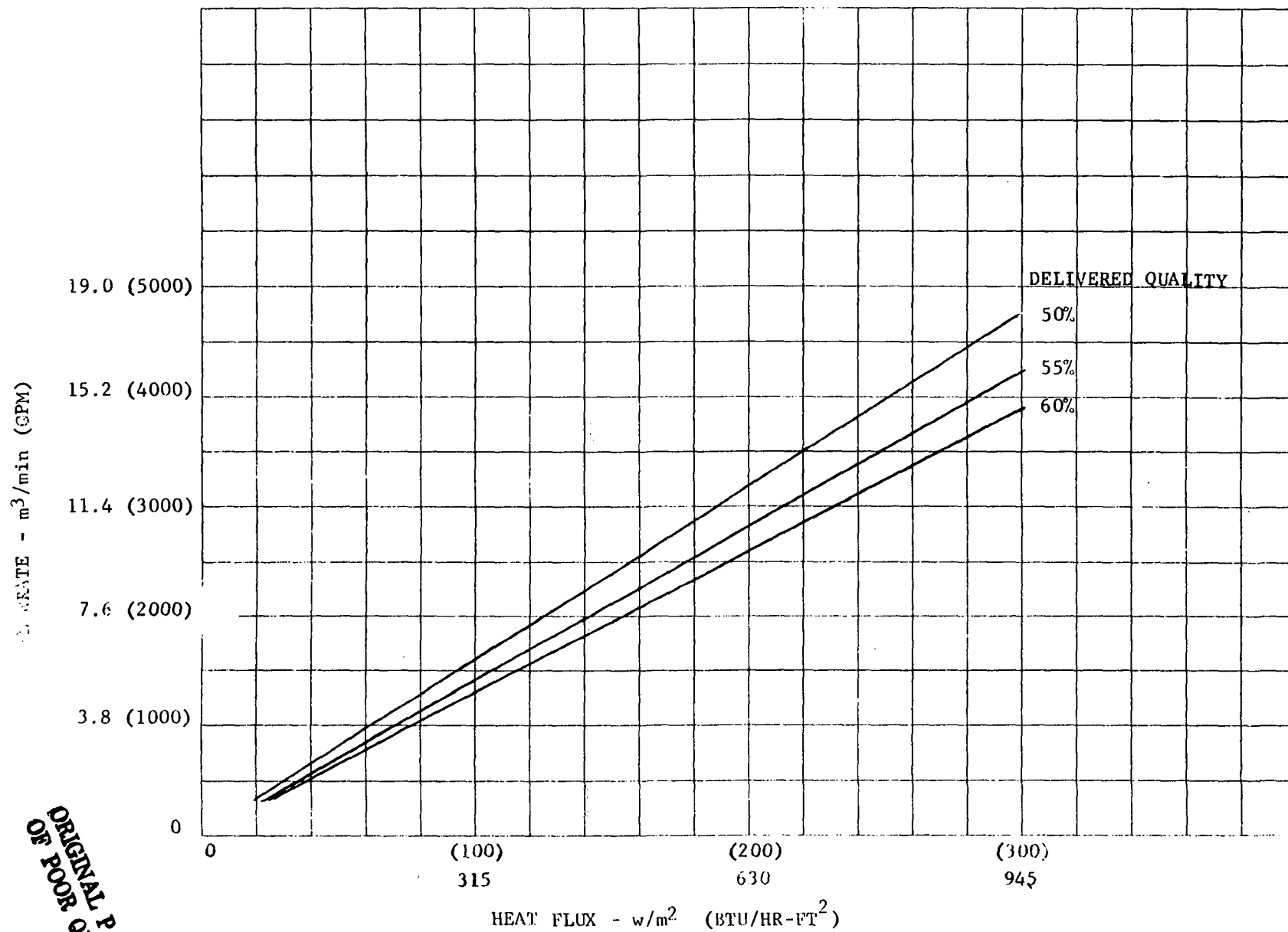


FIGURE 11-13

SSTO SLH₂ TANK - PAD HOLD FLOWRATE VS HEAT FLUX

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outflow and the vehicle tank fill models; and the flight load, loading rates and times established above. Through numerous iterations of the storage tank and vehicle tank models, the loading sequences for the 20/25 cm (8/10-inch) and 30.5 cm (12-inch) systems were determined as presented in Table II-4. Figures II-14 and II-15 present the data from the storage and vehicle tank models for the 20/25 cm (8/10-inch) system while Figures II-16 and II-17 present the 30.5 cm (12-inch) system data.

For the 20/25 cm (8/10-inch) system, the entire loading operation including a 45-minute pad hold period is accomplished in 103 minutes with 2320.5 m³ (613,000 gal) [191,870 Kg (423,000 lb)] of 60% SLH₂ transferred from the storage tank. Also, 34,382 Kg (75,800 lb) of 60% SLH₂ were converted to liquid at 22.6°K (40.6 °R) due to pressurization resulting in an enthalpy loss of 112 KJ/Kg (48.4 Btu/lb) or 3.87 x 10⁶ KJ (3.67 x 10⁶) Btu added to the storage tank propellant.

For the 12-inch system, the total time is 95 minutes with 2138.75 m³ (565,000 gal) [176,900 Kg (390,000 lb)] of 60% SLH₂ transferred. Pressurization resulted in a 30,300 Kg (66,800 lb) stratified layer at 20.7°K (37.2°R) for an enthalpy loss of 91.4 KJ/kg (39.3 Btu/lb) or 2.76 x 10⁶ KJ (2.62 x 10⁶ Btu) into the residual tank propellant.

Therefore, by modifying the existing system to 30.5 cm (12-inch) line size, a savings of 182 m³ (48,000 gal) of 60% SLH₂ and 8 minutes per launch are realized. Also, the heat transferred into the residual storage tank propellant is reduced by 1 x 10⁶ KJ (1 x 10⁶) Btu.

- d. Storage Tank Capacity - The quantity of 60% SLH₂ and of total propellant remaining in the storage tank at any time during the loading is plotted in Figures II-14 and II-16. The minimum tank capacity required to support a loading is that which insures the 60% SLH₂ is not depleted before loading terminates. After vehicle loading and pad hold for 45 minutes with the 20/25 cm (8/10-inch) system only 6,800 Kg (15,000 lb) of the initial 227,700 Kg (65,000 lb) remain at termination of the loading sequence. Therefore, the existing tanks will support SLH₂ loadings with either size system but longer pad hold times would require larger tanks.
- e. Storage Tank Pressurization - The storage tank ullage pressure profile is shown in Figures II-14 and II-16 for the 20/25 cm

TABLE II-4 SSTO/50% SLH₂ LOADING SEQUENCE

SYSTEM	OPERATION	% LOAD	TIME min	FLOWRATE m ³ /min (GPM)		QUANTITY OF 60% SLH ₂ TRANSFERRED m ³ (GAL.)		QUALITY DELIVERED TO VEHICLE (% SOLID)	STORAGE TANK ULLAGE PRESSURE KPa (PSIA)	
Existing 20 and 25 cm (8 and 10- inch)	Line Chillo down & Storage Tank Pressurization	0	6.5	0-5.9	(0-1560)	4.92	(1,300)	0-58.5	0-448	(0-65)
	Initial Fill	0-2	5.0	5.9	(1,560)	29.53	(7,800)	58.5	448	(65)
	Fast Fill	2-100	37.7	37.85	(10,000)	1427	(377,000)	53.5	448	(65)
	Upgrading	100	9.2	37.85	(10,000)	348.3	(92,000)	53.5	448	(65)
	Pad Hold	100	45	11.36	(3,000)	511.0	(135,000)	58.3	Decay 207	to (30)
	TOTALS	100	103.4			2321	(613,100)			
30.5 cm (12-inch)	Line Chillo down & Storage Tank Pressurization	0	6.5	0-5.9	(0-1560)	6.81	(1,800)	0-58.5	0-290	(0-42)
	Initial Fill	0-2	5.0	5.9	(1,560)	29.53	(7,800)	57.0	290	(42)
	Fast Fill	2-100	34.3	41.6	(11,000)	1427	(377,000)	57.0	290	(42)
	Upgrading	100	4.0	41.6	(11,000)	166.6	(44,000)	57.0	290	(42)
	Pad Hold	100	45	11.3	(2,990)	509.1	(134,500)	58.9	Decay 207	to (30)
	TOTAL	100	94.8			2139	(565,100)			

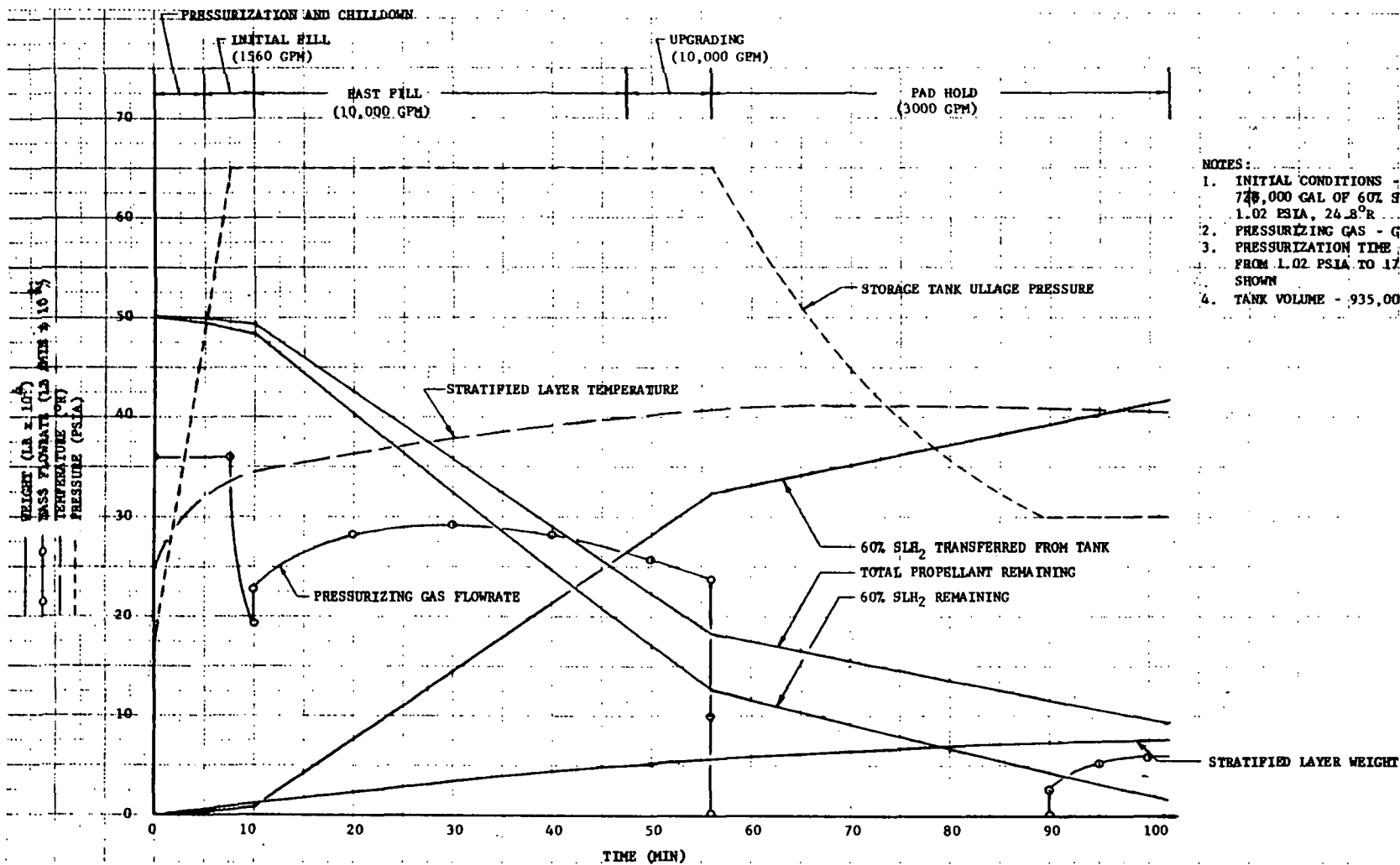
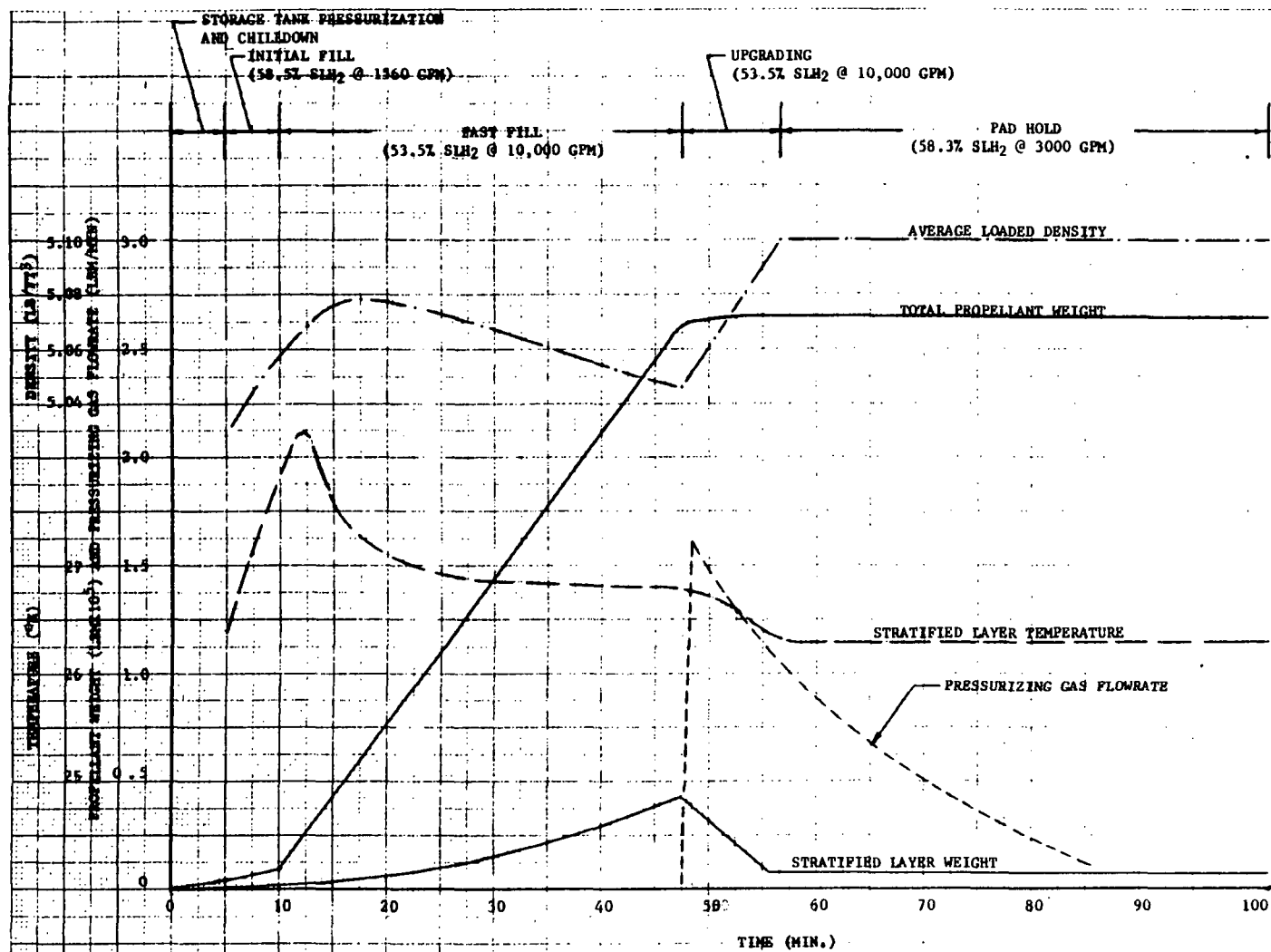


FIGURE II-14

SSO/SLH₂ PROPELLANT LOADING TIMELINE - STORAGE TANK PROFILE FOR EXISTING 8 AND 10 INCH TRANSFER SYSTEM



NOTES:

1. TANK INSULATION - 1 INCH
INTERNAL PFO FOAM
(225 BTU/HR·ft²)
2. INITIAL TANK TEMP - 530°R
3. PRESSURIZING GAS - GHD @ 530°R
4. TANK PRESSURE MAINTAINED AT
15 PSIA

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FIGURE II-15

SSTO/SLH₂ PROPELLANT LOADING TIMELINE - VEHICLE TANK PROFILE FOR EXISTING 8 & 10" TRANSFER SYSTEM

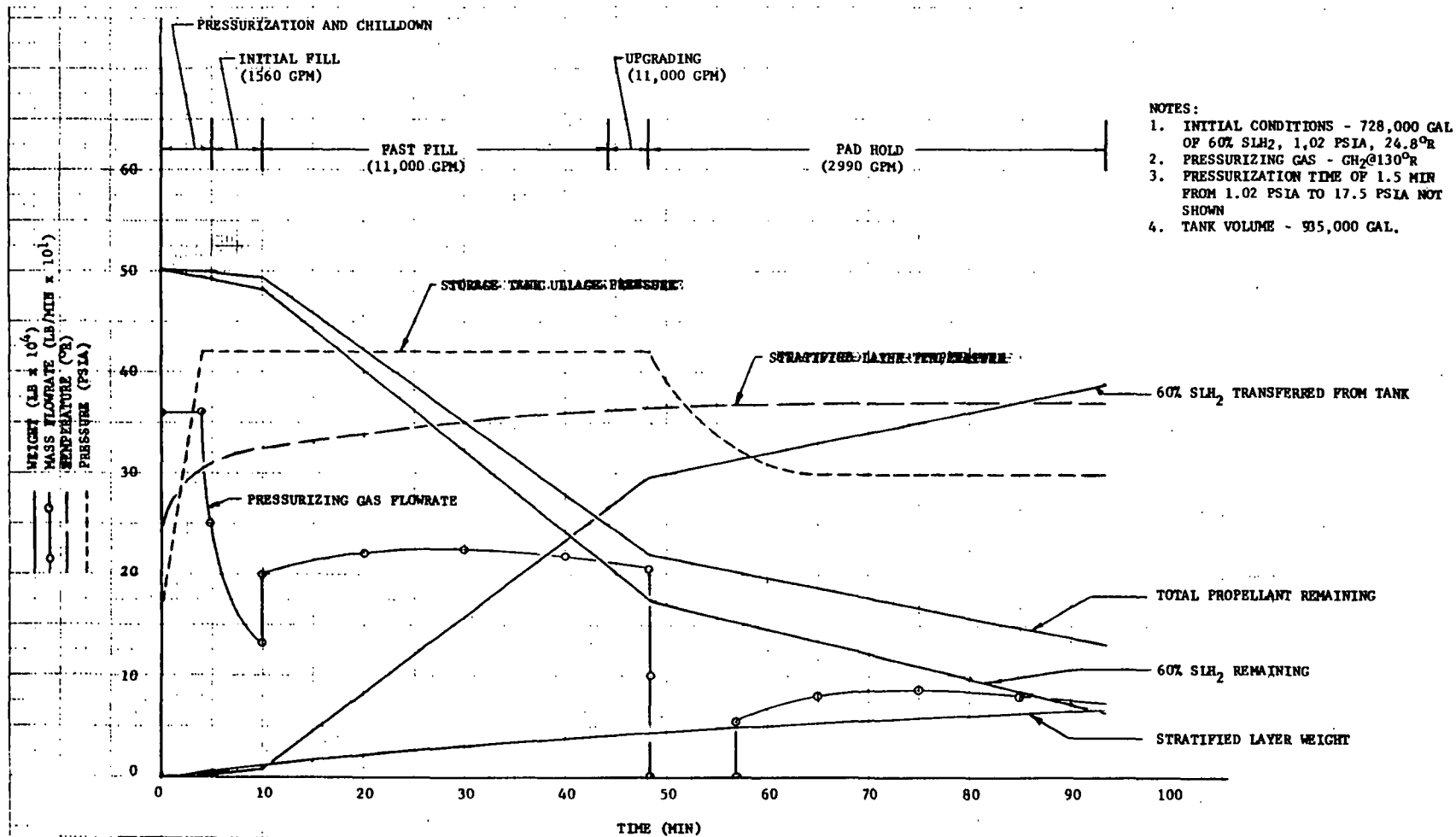
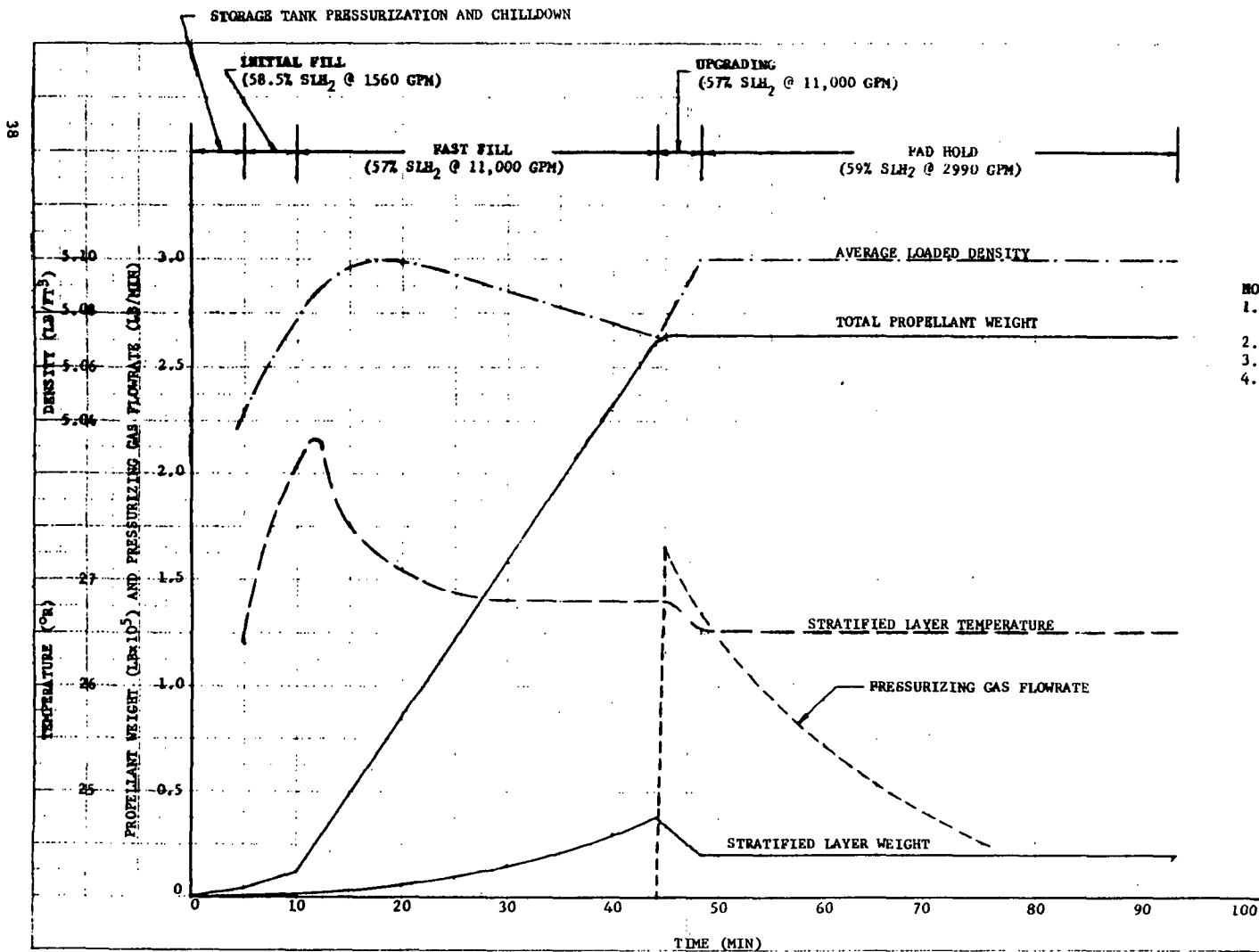


FIGURE II-16

SSTO/SLH₂ PROPELLANT LOADING TIMELINE - STORAGE TANK PROFILE FOR 12 INCH TRANSFER SYSTEM

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NOTES:

1. TANK INSULATION - 1 INCH INTERNAL PPO FOAM (225 BTU/HR-FT²)
2. INITIAL TANK TEMP - 530°R
3. PRESSURIZING GAS - GHE @ 530 R
4. TANK PRESSURE MAINTAINED AT 15 PSIA

FIGURE II - 17

SSTO/SLH₂ PROPELLANT LOADING TIMELINE - VEHICLE TANK PROFILE FOR 12th TRANSFER SYSTEM

(8/10-inch) and 30.5 cm (12-inch) systems, respectively. For the 20/25 cm (8/10-inch) system, a pressure of 448 KPa (65 psia) is required to sustain the 37.85 m³/min (10,000 gpm) fast fill rate and overcome the 59 KPa (8.5 psi) elevation head. At the termination of upgrading the SLH₂ flow to the storage tank vaporizer is shut off. This allows the storage tank pressure to decay to 207 KPa (30 psia) at which point the vaporizer supply valve throttles open to regulate the tank pressure constant at 207 KPa (30 psia).

For the 30.5 cm (12-inch) system, an ullage pressure of 290 KPa (42 psia) is required for the fast fill flowrate of 41.6 m³/min (11,000 gpm) and also decays to 207 KPa (30 psia) for the pad hold sequence.

The existing storage tanks at LC 39 were designed for a maximum operating pressure of 621 KPa (90 psia) (Reference 6). Therefore, a pressurized transfer of 60% SLH₂ in the existing 20/25 cm (8/10-inch) or a 30.5 cm (12-inch) system would cause no structural problem.

The flowrate of GH₂ pressurant gas required to maintain tank pressure is also shown in Figures II-14 and II-16. For the 20/25 cm (8/10-inch) system 7.5 minutes of 163 Kg/min (360 lb/min) GH₂ is required for initial pressurization to 448 KPa (65 psia). To maintain pressure during fast fill outflow, a maximum of 132 Kg/min (290 lb/min) is required. If the initial pressurization time were extended, the vaporizer flow requirement could be reduced to 132 Kg/min (290 lb/min).

The existing LC 39 LH₂ system has two 81.6 Kg/min (180 lb/min) capacity vaporizers connected in parallel with the storage tank. The primary vaporizer flow control valve is actuated automatically by sensing tank ullage pressure and the redundant vaporizer supply valve operates manually. By utilizing both vaporizers simultaneously, the required flowrates for SLH₂ can be achieved at the expense of redundancy.

- f. Pressurized Discharge versus Pump Transfer - As previously discussed, most LH₂ systems including those at KSC transfer by pressurized expulsion due to its low density and relative simplicity of the system. Since the heating from pressurization is significant, however, an analysis of pump transfer heating was conducted.

A comparison was made to the 30.5 cm (12-inch) system, pressurized transfer case requiring a discharge pressure of 290 KPa (42 psia)

at 41.6 m³/min (11,000 gpm). It was assumed that a pump similar to the 37.85 m³/min (10,000 gpm) LOX pump at LC 39 (Reference 17) would be used have an efficiency of 80% with a 186 KPa (27 psig) head pressure and 170 KPa (10 psig) NPSH. The pump heating was calculated using equations in Reference 13 for the 30.5 cm (12-inch) system loading flowrates and timeline defined in Table II-4. The resulting heat input from the pump alone was determined to be 106,000 KJ/launch (100,000 Btu/launch).

To provide the 170.3 KPa (10 psig) to the pump, however, the storage tank must be pressurized. The storage tank pressurization/stratification program was run at the 30.5 (12-inch) system loading rates but maintaining tank pressure at 170 KPa (10 psig). The resulting heat input calculated from the final stratified layer temperature and weight was 2×10^6 KJ (2×10^6 Btu). The total heating resulting from pumping [2.2×10^6 KJ (2.1×10^6 Btu)] is less than the 2.8×10^6 KJ (2.7×10^6 Btu) for the pressurized expulsion case.

The analysis does not consider such factors as pump motor power, heating from pump chardown and recirculation loops and pump system design and operational complexity which tends to offset this difference. For this study, therefore, further analyses will assume pressurized expulsion. For design of future SLH₂ systems, however, pump transfer should definitely be considered and development of low NPSH SLH₂ pumps pursued.

- g. Vehicle Tank Pressurization - The vehicle tank pressurization requirements, as determined by the pressurization/stratification program, are shown in Figures II-15 and II-17 for the 20/25 cm (8/10-inch and 30.5 cm (12-inch) loading systems, respectively.

It is interesting to note that no pressurization is required until outflow (upgrading) begins after the fast fill sequence. Prior to this time, the propellant boiloff rate exceeds the ullage gas condensation rate with the excess boiloff propellant exhausted out the vehicle vent. At the transition from fast fill to upgrading, the vehicle vent valve closes and pressurization begins at its maximum rate of .73 Kg/min (1.6 lb/min) for the 20/25 cm (8/10-inch) system and .77 Kg/min (1.7 lb/min) for the 30.5 cm (12-inch) system due to the slightly colder stratified layer temperature. Integration of the flowrate versus time curve results in the total GHe consumption of approximately 12 Kg (27 lb) for both systems.

- h. SSTO/SLH₂ Consumption Rate - The rate at which SLH₂ is consumed or must be produced to support SSTO flights is the sum of the vehicle flight load plus that required to make up for storage and transfer losses. The quantity required to make up for losses is calculated from total mass and enthalpy associated with each heat source and is expressed as refrigeration loss, i.e., tons/day or tons per launch. This refrigeration loss for both the 20/25 cm (8/10-inch) and 30.5 cm (12-inch) systems is tabulated in Table II-5 and is shown graphically in Figure II-18.

In Figure II-18 it is noted that for launch rates less than 3 per year, the storage tank heat leak is the major source of enthalpy gain. Above this rate, the heat input from the vehicle tank and storage tank pressurization are by far the major contributors to enthalpy gain.

Table II-5. SLH₂ Storage and Transfer Refrigeration Loss

Heat Source	Units	20/25 cm (8/10-inch) System	30.5 cm (12-inch) System
Storage tank heat leak	Kg/day (Tons/day)	275 (0.303)	275 (0.303)
Transfer Line Chiltdown	Kg/launch (Tons/launch)	400 (0.44)	550 (0.61)
Storage Tank pressurization	Kg/launch (Tons/launch)	34,400 (37.9)	30,300 (33.4)
Transfer line friction & environment	Kg/launch (Tons/launch)	3,330 (3.67)	1,420 (1.56)
Vehicle tank chiltdown, heat leak & pressuriza- tion	Kg/launch (Tons/launch)	42,200 (46.5)	39,800 (43.9)

Figure II-19 is a plot of total 60% SLH₂ consumption rate for both size systems as a function of SSTO launch rate. The 60% SLH₂ production capacity required to support the anticipated SSTO launch rate of 24 to 140 launches per year (Reference 1)

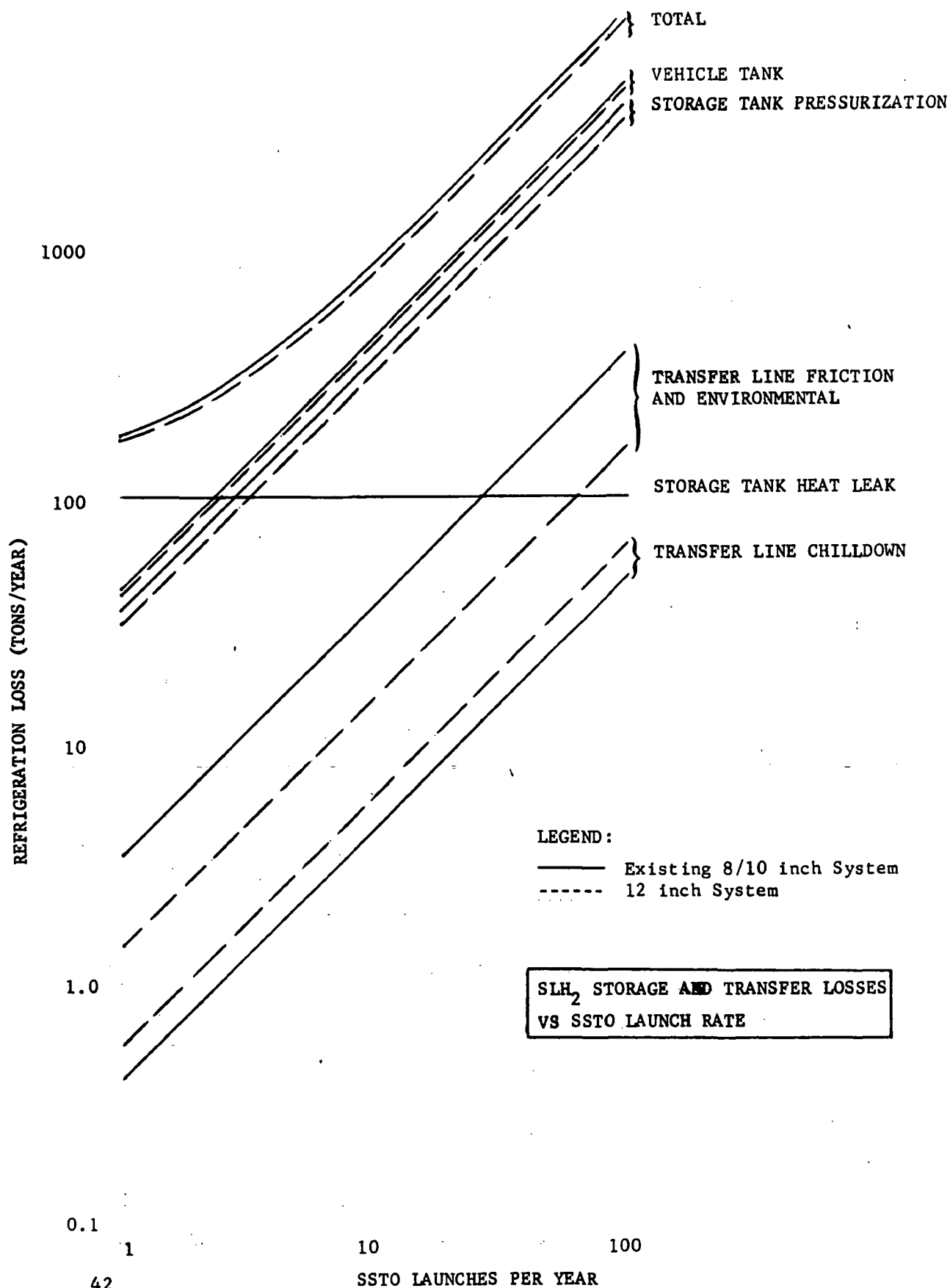


FIGURE II-18

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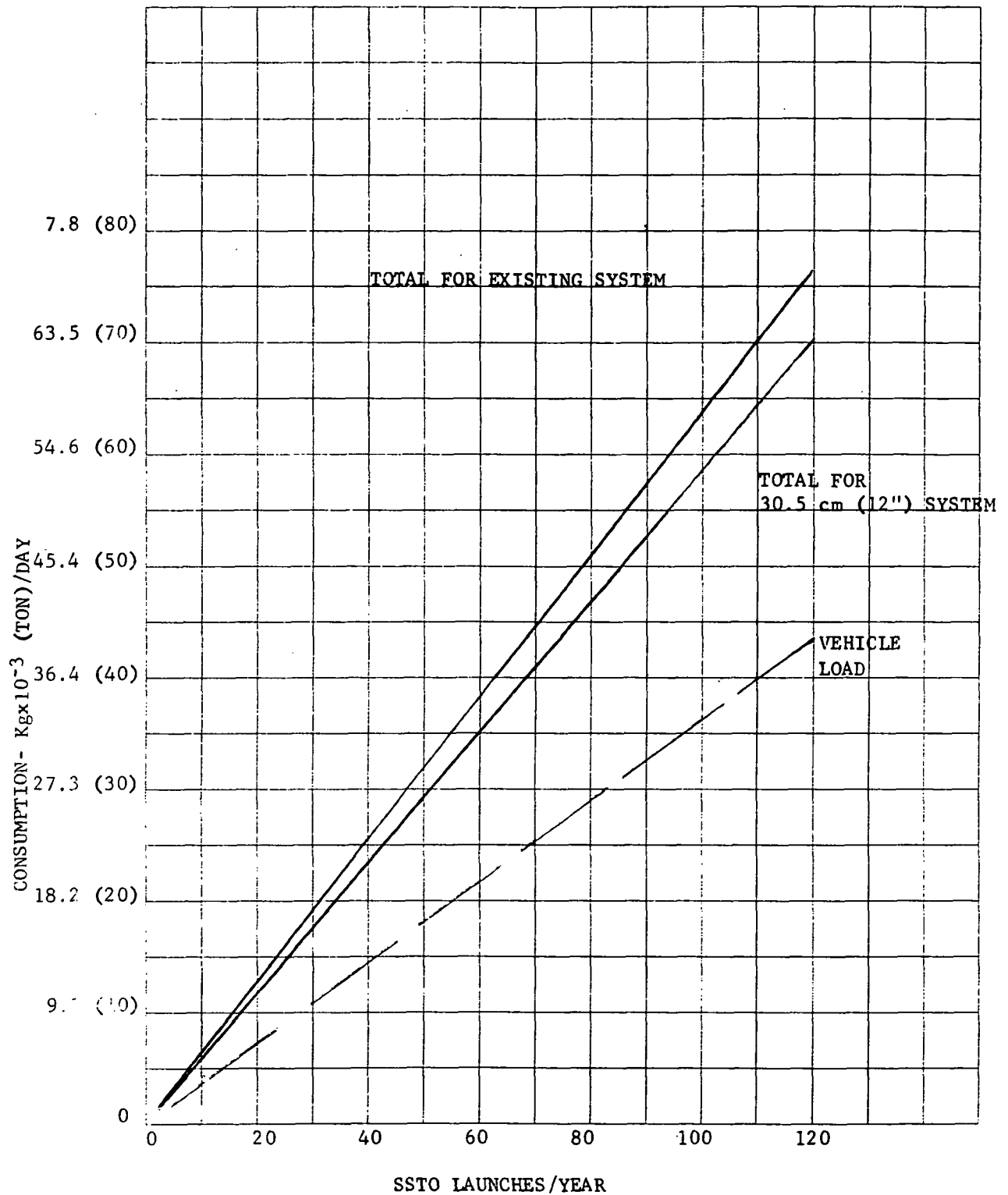


FIGURE II-19 60% SLH₂ CONSUMPTION VS SSTS LAUNCH RATE

is 51,000 and 284,000 Kg/day (16 and 89 tons/day), respectively, for the existing 20/25 cm (8/10-inch) system. For a 30.5 cm (12-inch) system this reduces to 12,700 and 261,000 Kg/day (14 and 82 tons/day) for a net savings of 2800 to 6400 Kg/day (2 to 7 tons) per day. A 13600 Kg/day (15 ton/day) 60% SLH₂ production plant as prescribed by the Statement of Work for² this contract will support 23 launches per year with the existing 20/25 cm (8/10-inch) system and 25 launches/ year with a 30.5 (12-inch) system.

B. TRIPLE POINT LIQUID HYDROGEN STORAGE AND TRANSFER

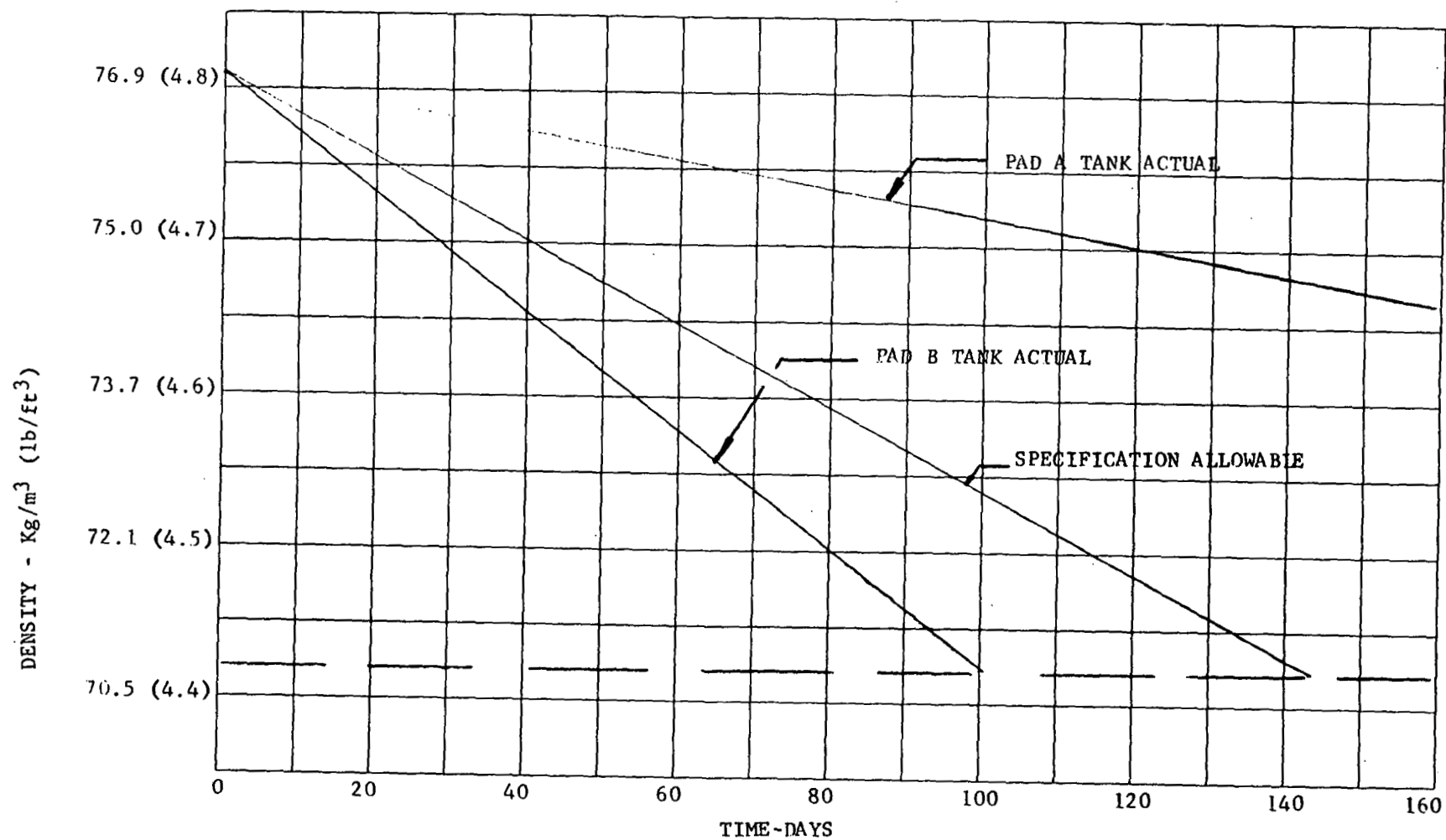
The triple point liquid hydrogen (TP LH₂) analyses were conducted with a set of baseline requirements which included an SSTO vehicle (References 1, 2 and 3) utilizing TP LH₂ and TP LOX at a MR of 6:1 as propellants (see Figure II-1). This vehicle has a GLOW of 1,146,308 Kg (2,527,176 lb) and TP LH₂ capacity of 123,656 Kg (272,615 lb) or 1605 m³ (424,000 gal) at a density of 77 Kg/m³ (4.81 lb/ft³).

As was the case for the SLH₂ analysis, the existing Shuttle LH₂ loading system at KSC LC 39² (References 4, 5, and 6) is used as the baseline for the ground system analysis (see Figure II-2). The loading sequence for Shuttle (see Table II-1) serves as a basis for the loading timeline analysis. The iteration process, as described in detail for the SLH₂ system, was repeated for TP LH₂ to evaluate the storage tank, transfer line and vehicle tank enthalpy gain and to establish the optimum loading sequence. therefore, only the analysis results for the TP LH₂ system will be discussed in this section.

1. Storage Tank

The existing LC 39 LH₂ storage tanks have a maximum capacity of 3217.6 m³ (850,000 gal) of LH₂ with a corresponding weight of 277,700 Kg (502,000 lb). To maintain the same weight, the maximum capacity of TP LH₂ is 2952.6 m³ (780,000 gal) due to its higher density 77 versus 70.8 Kg/m³ (4.81 versus 4.42 lb/ft³). For a detailed description of this tank see Para. I.A.1 and Reference 6. To determine if this configuration tank is suitable for use with TP LH₂ for SSTO the standby heat leak, standby pressure and its pressurization and stratification during loaidn were studied.

- a. Storage Tank Heat Leak - The average density decay in these tanks initially loaded with TP LH₂ is shown in Figure II-20. After 15 days, corresponding to the minimum SSTO launch rate, the density in the Pad A tank decays to 76.8 Kg/m³ (4.795 lb/ft³) while the



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FIGURE II-20 TRIPLE POINT LH₂ DENSITY DECAY IN LC 39 LH₂ STORAGE TANKS

Pad B tank decays to 76.1 Kg/m^3 (4.75 lb/ft^3).

- b. Storage Tank Standby Pressure - The existing tanks are capable of withstanding a full vacuum in the inner tank. Therefore, the storage of TP LH_2 at its equilibrium vapor pressure of 7.03 KPa (1.02 psia) will cause no structural problem.

The effect on the density and temperature of subcooled hydrogen stored in the specification LC 39 storage tank at triple point pressure versus 1 atmosphere pressure is shown in Figure II-21 and II-22. The tank specification boiloff rate of 0.075%/day at 20.6°K (37°R) and 305.6°K (550°R) ambient (Reference 6) was equated to heat leak rate of 909w ($74,461 \text{ Btu/day}$) at triple point temperature with the tank full at 3217.6 m^3 ($850,000 \text{ gal}$). This amount of refrigeration would be required to maintain the density and temperature at triple point conditions.

It was assumed that the ullage gas is in thermal equilibrium with the bulk liquid. This would normally be the case if the propellant were initially stored at 7.03 KPa (1.02 psia) and allowed to slowly self-pressurize from the heat input through the tank walls. However, in the case of 101 KPa (14.7 psia) storage, if the propellant were pressurized to 101 KPa (14.7 psia) by an external source, heat transfer into the liquid would occur if the pressurant gas was not at 13.9°K (25°R). For this analysis, it was assumed that this pressurization to 101 KPa (14.7 psia) occurred elsewhere and was in equilibrium when delivered to the storage tank.

In Figure II-21 it is shown that the density is slightly higher for the propellant stored at 1 atm up to approximately 75 days of storage and is due to the higher pressure. The rate of density decay is greater for the propellant stored at 1 atm due to the heat of compression and after 75 days the propellant initially stored at 7.03 KPa (1.02 psia) is of higher density. In Figure II-22, it is shown that the subcooled hydrogen maintained at 101 KPa (14.7 psia) is slightly warmer and has a greater rate of temperature rise, again due to the heat of compression.

The filling of the storage tank and the transfer during vehicle loading require system pressures above one atmosphere. Therefore, since system operation is above atmospheric pressure and since storage at one atmosphere does not have an appreciable effect on the temperature and density, there is no need to store at triple point pressure.

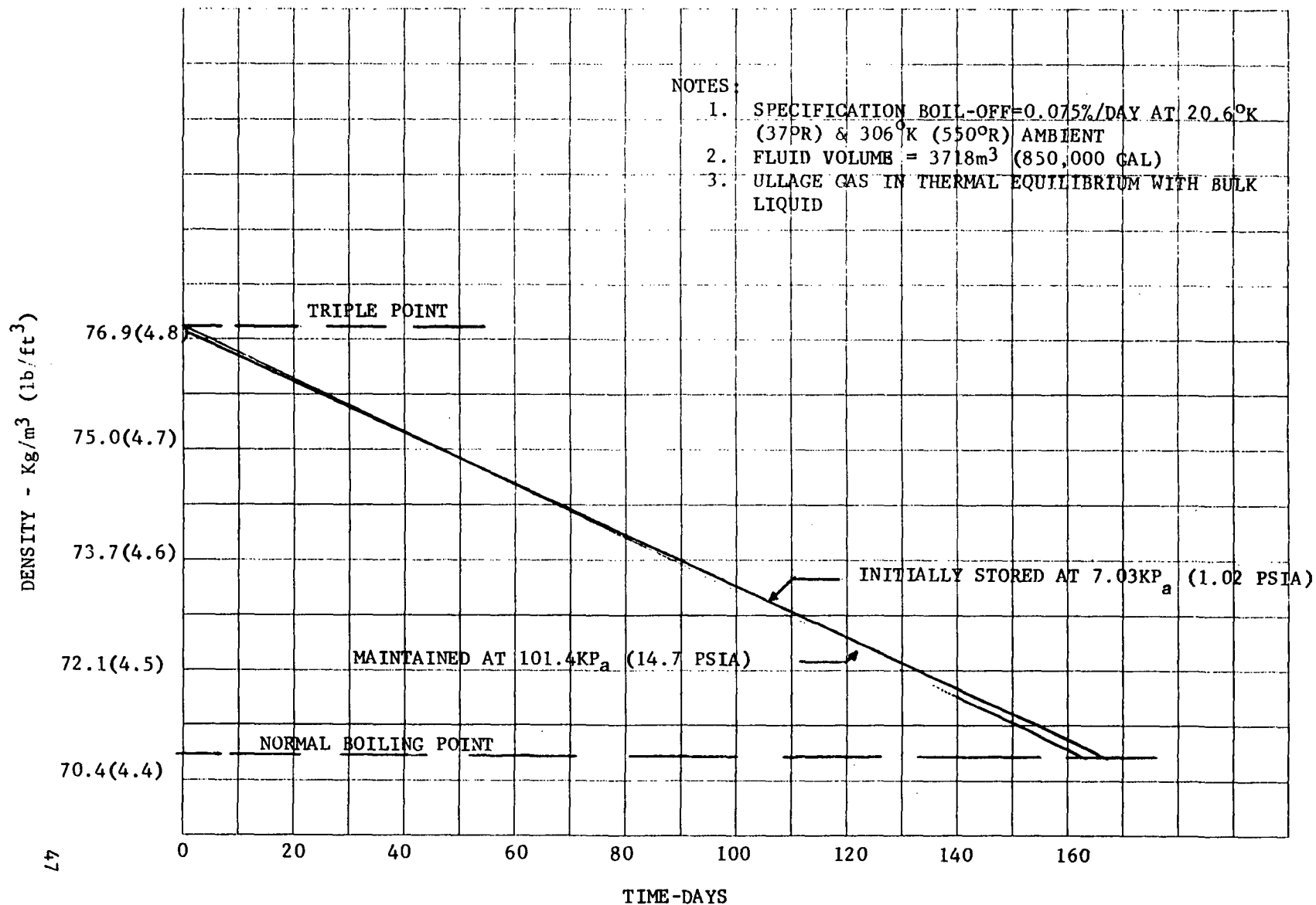


FIGURE II-21 SUBCOOLED LH₂ AVERAGE DENSITY DECAY IN LC 39 STORAGE TANKS

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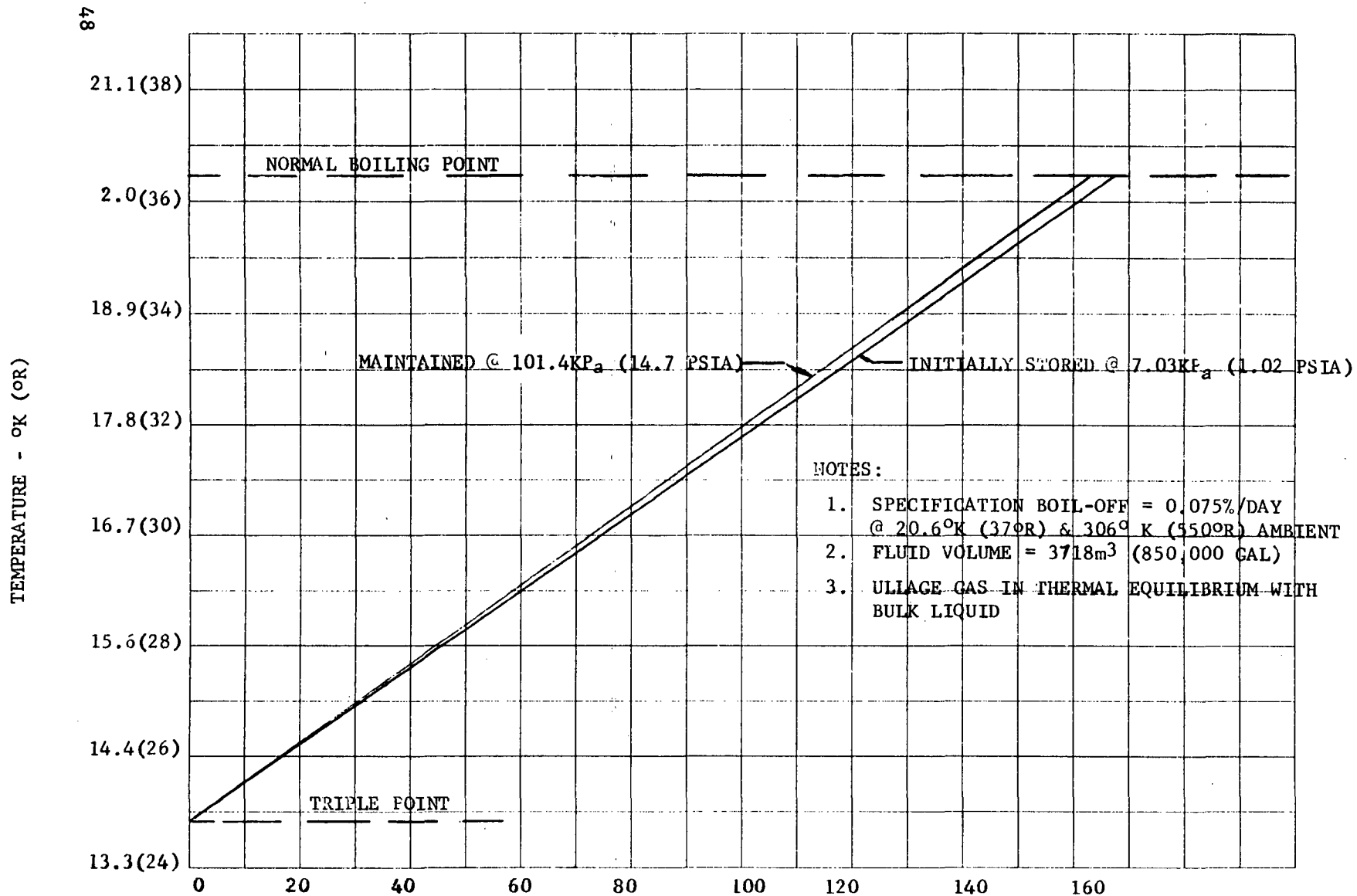


FIGURE II-22 SUB-COOLED LH₂ AVERAGE TEMPERATURE RISE IN LC 39 LH₂ STORAGE TANKS

Therefore, it is recommended that the TP LH₂ storage tank be designed to allow for triple point pressure storage, but the tank pressure would normally be at or above one atmosphere.

- c. Storage Tank Pressurization and Stratification - The SLUSHPRESS computer program described in Appendix A was utilized for the storage tank analysis of TP LH₂. The model assumes the tank is initially filled with a homogeneous mixture of subcooled hydrogen at triple point temperature. The heat from pressurization forms a stratified layer of warmer liquid on top of the bulk mass of propellant at triple point temperature resulting in the outflow of high density propellant from the bottom of the tank.

To develop a parametric relationship between storage tank pressure during loading, several computer runs were made with the pressure and flowrate data in Table II-6. The storage tank pressure required for transfer is the summation of the transfer line pressure drop and the TP LH₂ elevation head between the storage tank and the fully loaded SSTO tank. The line pressure drop for varying flowrates and line sizes is discussed in Para I.B.3. The elevation head, shown in Figure II-2, is 73.8m (242-Ft) of TP LH₂ or approximately 56 KPa (8.1 psi).

Table II-6 TP LH₂ Storage Tank Heat Input

Transfer Line size cm (Inch)	Fast Fill Flowrate m ³ /min (gpm)	Storage Tank Ullage Press. KPa (psia)	Heat Input KJx10 ⁶ (Btux10 ⁶)
20 x 25 8 x 10	36.0 9,500	414 60	6.40 6.07
25 10	37.9 10,000	386 56	5.59 5.30
30.5 12	39.7 10,500	276 40	4.33 4.11
35.5 14	43.5 11,500	255 37	3.44 3.26

The results are plotted with the SLH₂ system data in Figure II-7 and show the weight and temperature of the stratified layer at the end of the SSTO loading sequence as a function of transfer line size.

Since the tank is full of TP LH₂ initially, the heat input from pressurization can be calculated by subtracting the final enthalpy of the stratified layer from the enthalpy of TP LH₂ [307.7KJ/Kg (132.3 Btu/lb)] and multiplying by the stratified layer weight.

2. Transfer Line

The existing LC 39 transfer line is a multilayer insulated vacuum jacketed pipe 518 m (1700 ft) of which 25 cm (10-inch) diameter and 15 m (50 ft) is 20 cm (8-inch) diameter. Refer to Para. I. A.2 and Reference 6 for more specific information of the line design.

The sources of propellant heating between the storage tank and vehicle tank are the transfer line friction, environmental heating and chilldown and were determined as previously described for SLH₂.

The transfer line pressure drop used to calculate the friction heat input is essentially the same for TP LH₂ as for SLH₂ and is shown in Figure II-8. This is due to the fact that the filter pressure drop was not included for the SLH₂ line and the increase in ΔP from the TP LH₂ filter is essentially equal to the increase in SLH₂ line ΔP resulting from its higher density.

A detailed thermal analysis of the transfer line for TP LH₂ is included in Appendix B. The results show a vacuum jacketed line with MLI, similar to the existing LC 39 design is the best choice for subcooled hydrogen. The average environmental heat leak was established at 9.6 w/m (10 Btu/hr/ft of line) for the 25 cm (10-inch) diameter line size.

With the line pressure drop and heat leak defined, the steady-state enthalpy rise resulting from friction and environmental heating can be determined. The increase in enthalpy due to transfer line piping and component friction can be calculated from equation (4).

$$\Delta H_{\text{friction}} = \frac{\Delta P}{\rho} \left(\frac{144}{778} \frac{\Delta P}{\rho} \right) \quad (4)$$

where ΔH = change in enthalpy (Btu/lb)
 ΔP = friction pressure drop (psi)
 ρ = density (lb/ft³)

The enthalpy rise due to transfer line environmental heating is determined by the equation (5).

$$\Delta H_{\text{env}} = \frac{qL}{V\rho} \quad (5)$$

where: q = transfer line heat leak (Btu/hr-ft)
 L = length of line (ft)
 V = volumetric flowrate (ft³/hr)

The summation of these two expressions is the change in enthalpy between the storage tank and the vehicle tank and is plotted in Figure II-23 as a function of flowrate and line size. The net system heat input is determined by multiplying ΔH by mass flowrate and transfer duration.

For the 20/25 cm (8/10-inch) system, the minimum enthalpy gain is 9.3 KJ/Kg (4 Btu/lb) at 7.57 m³/min (2000 gpm). With triple point propellant delivered from the storage tank, the enthalpy of propellant entering the vehicle tank, at a storage tank pressure of 207 KPa (30 psia), is $-306.3 + 9.3 = -297.0$ KJ/Kg ($-131.7 + 4.0 = -127.7$ Btu/lb). This corresponds to a delivered density of 75.72 Kg/m³ (4.727 lb/ft³) and is the maximum density which the existing 20/25 cm (8/10-inch) system can deliver to the vehicle. For a 20/25 cm (12-inch) system, the minimum enthalpy 6.5 KJ/Kg (2.8 Btu/lb) at 13.25 m³/min (3500 gpm) and corresponds to a maximum deliverable density of 76.07 Kg/m³ (4.749 lb/ft³).

The heat transferred to the propellant for line chilldown is the same as for the SLH₂ system and is 1.8×10^6 KJ (1.7×10^6 Btu) for the existing system.

3. Vehicle Tank

The fuel tank of the baseline SSTO (TP LH₂/TP LOX) vehicle as defined by Reference 1, has a volume of 1654 m³ (58,400 ft³) and contains 123,656 Kg (272,615 lb) [1605.3 m³ (424,085 gal)] of propellant at liftoff. The heat transfer area of the tank is 926.9 m² (9977 ft²). A detailed analysis of the vehicle insulation system is included as Appendix C and recommends 2.5 cm (1.0 inch) internal PPO foam insulation having an average heat flux of 709 w/m² (225 Btu/ft²/hr). The total heat input to the propellant in a PPO foam-insulated SSTO tank is shown in Figure II-10 as a function of insulation thickness and total loading time. This curve includes heat input from the chilldown and environmental heat leak.

For a vehicle tank load of 1605.0 m³ (424,000 gal) of TP LH₂ and a constant 45-minute pad hold period, the total loading time and heat input can be determined as a function of fast fill flowrate as shown in Table II-7.

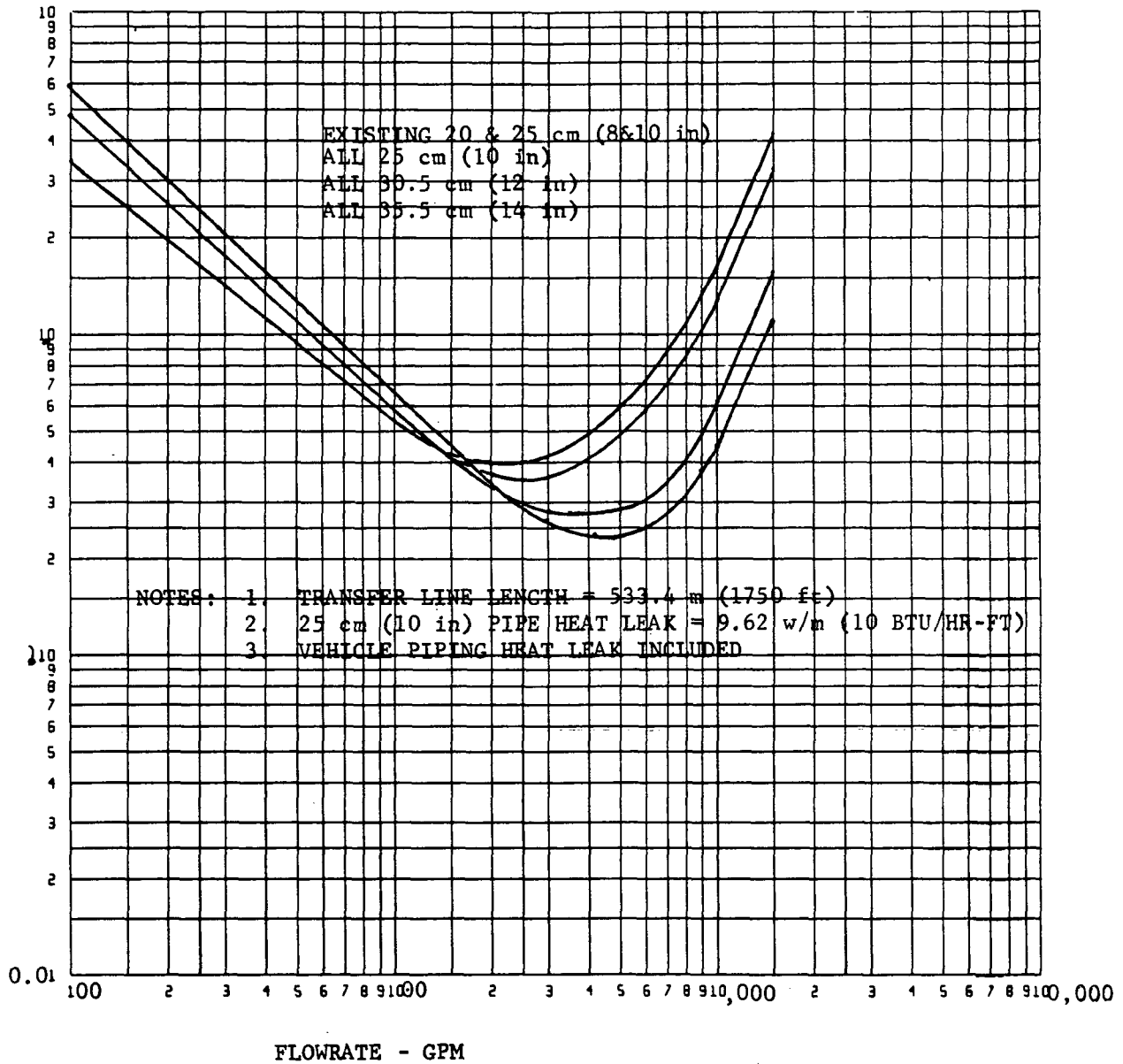


FIGURE II-23

TP LH₂ ENTHALPY DEGRADATION VS FLOWRATE- LC 39
 FRICTION AND ENVIRONMENTAL HEATING FOR VARYING
 TRANSFER LINE SIZES

Table II-7. TP LH₂ Vehicle Tank Heat Input (1-inch) PPO Foam

Fast Fill Rate m ³ /min (gpm)	Total Loading Time (min)	Heat Input KJ x 10 ⁶ (Btu x 10 ⁶)	
22.7 (6,000)	116	3.95	(3.75)
30.3 (8,000)	98	3.64	(3.45)
37.9 (10,000)	88	3.37	(3.20)
45.4 (12,000)	80	3.22	(3.05)

In order to evaluate the pressurization and stratification of TP LH₂ in the vehicle tank, the SLUSHPRESS model was input with appropriate TP LH₂ vehicle tank properties.

4. Total TP LH₂ System Analysis

- a. Fast Fill Transfer Rate - The total system heat input from the ground transfer system and the vehicle tank is shown in Figure II-24 as a function of flowrate for varying size transfer lines and varying insulation thickness. The total system heat input for a 2.5 cm (1-inch) PPO foam insulated vehicle tank is shown in Figure II-25 for the varying size transfer lines. The optimum fast fill flowrates determined from the point of minimum heat input are 36.0 m³/min and 39.74 m³/min (9500 gpm and 10,500 gpm) for the 20/25 cm (8/10-inch) and 30.5 cm (12-inch) systems, respectively.
- b. Pad Hold Transfer Rate - The flowrate of propellant required to maintain the average density achieved after loading to 100% is a function of the average on-board density, the delivered propellant density and the vehicle tank insulation. Several iterations of the vehicle tank model of the pressurization/stratification program were run to determine the steady-state pad hold flowrate for the 2.5 cm (1-inch) PPO foam insulated SSTO fuel tank. For the 20/25 cm (8/10-inch) system, a flowrate of 36.0 m³/min (9500 gpm) is required to maintain the loaded average density at 75.56 Kg/m³ (4.717 lb/ft³). A flowrate of 37.85 m³/min (10,000 gpm) and 14.1°K (25.4°R) delivered propellant will maintain 75.90 Kg/m³ (4.738 lb/ft³) average density with the 30.5 cm (12-inch) system. These high pad hold flowrates, as compared to the SLH₂ system are due to absence of the solid heat of fusion. From these steady-state pad hold flowrates, it is apparent

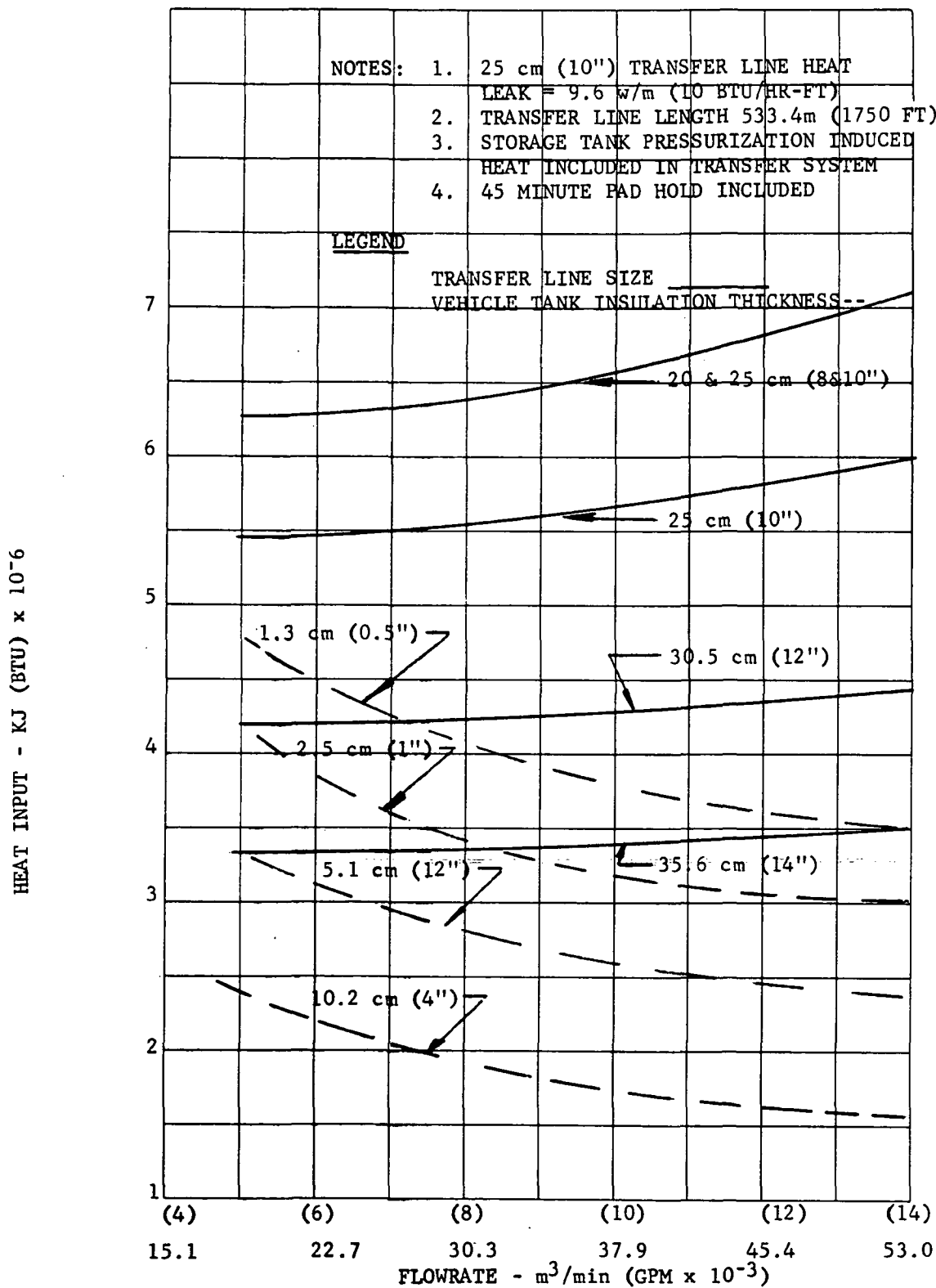


FIGURE II-24

SSTO/ TP LH₂ SYSTEM HEAT INPUT - VEHICLE TANK & TRANSFER SYSTEM HEAT VS FAST FILL FLOWRATE

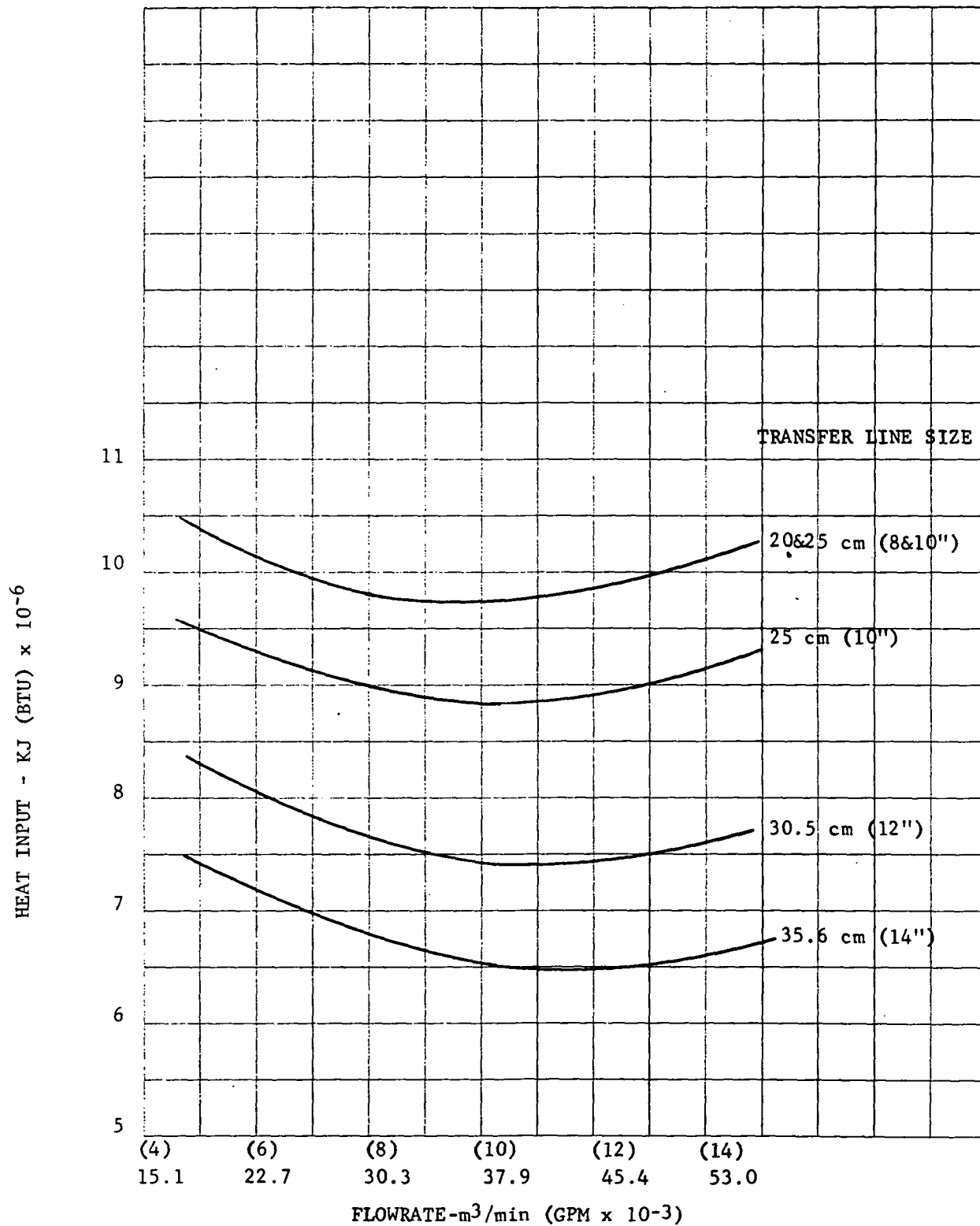


FIGURE II-25 SSTO/TP LH_2 SYSTEM HEAT INPUT - TOTAL HEAT VS FAST FILL
FLOWRATE FOR 2.54 cm (1 INCH) PPO FOAM INSULATED VEHICLE TANK

that to increase (upgrade) the average density in any reasonable amount of time, flowrate beyond practical limits would be required. Therefore, for the TP LH₂ analysis, the upgrading sequence is omitted and the pad hold sequence begins at fast fill termination.

- c. SSTO/TP LH₂ Loading Sequence - The loading sequence for the TP LH₂ fueled SSTO vehicle was determined by utilizing the storage and vehicle tank models of the pressurization/stratification program and the flight load, loading rates and times established above. Through numerous iterations of the two models, the loading sequences were determined as presented in Table II-8. Output data from the models are presented in Figures II-26 and II-27 for the 20/25 cm (8/10-inch) system and Figures II-28 and II-29 for the 30.5 cm (12-inch) system.

For the 20/25 cm (8/10-inch) system, the loading operation including a 45-minute pad hold period, requires 99 minutes with 3225 m³ (248,570 Kg) [852,000 gal (548,000 lb)] of TP LH₂ transferred from the storage tank to achieve an average liftoff density of 75.56 Kg/m³ (4.717 lb/ft³). Also, 88,450 Kg (195,000 lb) of liquid at triple point temperature were heated to 22.5°K (40.5°R) due to pressurization resulting in an enthalpy loss of 72.8 KJ/Kg (31.3 Btu/lb) or $(6.4 \times 10^6 \text{ KJ})$ into the tank propellant..

For the 30.5 cm (12-Inch) system, the loading time 95 minutes with 3316 m³ (255,370 Kg) [876,000 gal (563,000 lb)] of TP LH₂ transferred for a liftoff density of 75.89 Kg/m³ (4.738 lb/ft³). Tank pressurization resulted in a stratified layer of 83,000 Kg (183,000 lb) at 20.6°K (37°R) for an enthalpy loss of 55.4 KJ/Kg (23.8 Btu/lb) or $4.5 \times 10^6 \text{ KJ}$ ($4.3 \times 10^6 \text{ Btu}$) into the residual tank propellant. The larger quantity of total propellant transferred by the 30.5 cm (12-inch) system is due to the higher pad hold flowrate required to maintain the higher liftoff density. By increasing the existing system line size to 30.5 cm (12-inch), an increase in the average loaded density of 0.336 Kg/m³ (0.021 lb/ft³) is obtained.

Also, the heat transferred into the residual storage tank propellant is reduced by $1.9 \times 10^6 \text{ KJ}$ ($1.8 \times 10^6 \text{ Btu}$).

- d. Storage Tank Capacity - The initial iterations of the storage tank model revealed that the existing 3217.6 m³ (850,000 gal) capacity storage tank was not of sufficient size to deliver

TABLE II-8 SSTO/TP LH₂ LOADING SEQUENCE

SYSTEM	OPERATION	% LOAD	TIME (MIN.)	FLOWRATE m ³ /min (GPM)	QUANTITY OF TP LH ₂ TRANS. m ³ (GAL)	PROPELLANT TEMP. DELIVERED TO VEH. °K (°R)	STORAGE TANK PRESS. KPA (PSIA)
Existing 20 and 25 cm (8 and 10- Inch)	Line Chill- down & Storage Tank Press.	0	5	0-6.44 (0-1700)	4.92 (1,300)	13.8-14.0 (24.9-25.2)	0-414 (0-60)
	Initial Fill	0-2	5	6.44 (1700)	32.18 (8,500)	14.0 (25.2)	414 (60)
	Fast Fill	2-100	43.75	35.96 (9500)	1572.8 (415,500)	14.4 (25.9)	414 (60)
	Pad Hold	100	45	35.96 (9500)	1618.3 (427,500)	14.4 (25.9)	414 (60)
	TOTALS	100	98.75		3223.6 (851,585)		
30.5 cm (12-Inch)	Line Chill- down & Storage Tank Press.	0	5	0-6.44 (0-1700)	6.81 (1,800)	13.8-14.0 (24.9-25.2)	0-276 (0-40)
	Initial Fill	0-2	5	6.44 (1700)	6.81 (8,500)	14.0 (25.2)	276 (40)
	Fast Fill	2-100	39.58	39.75 (10,500)	1572.8 (415,500)	14.1 (25.4)	276 (40)
	Pad Hold	100	45	37.85 (10,000)	1703.4 (450,000)	14.1 (25.4)	276 (40)
	TOTALS	100	94.58		3314.1 (875,800)		

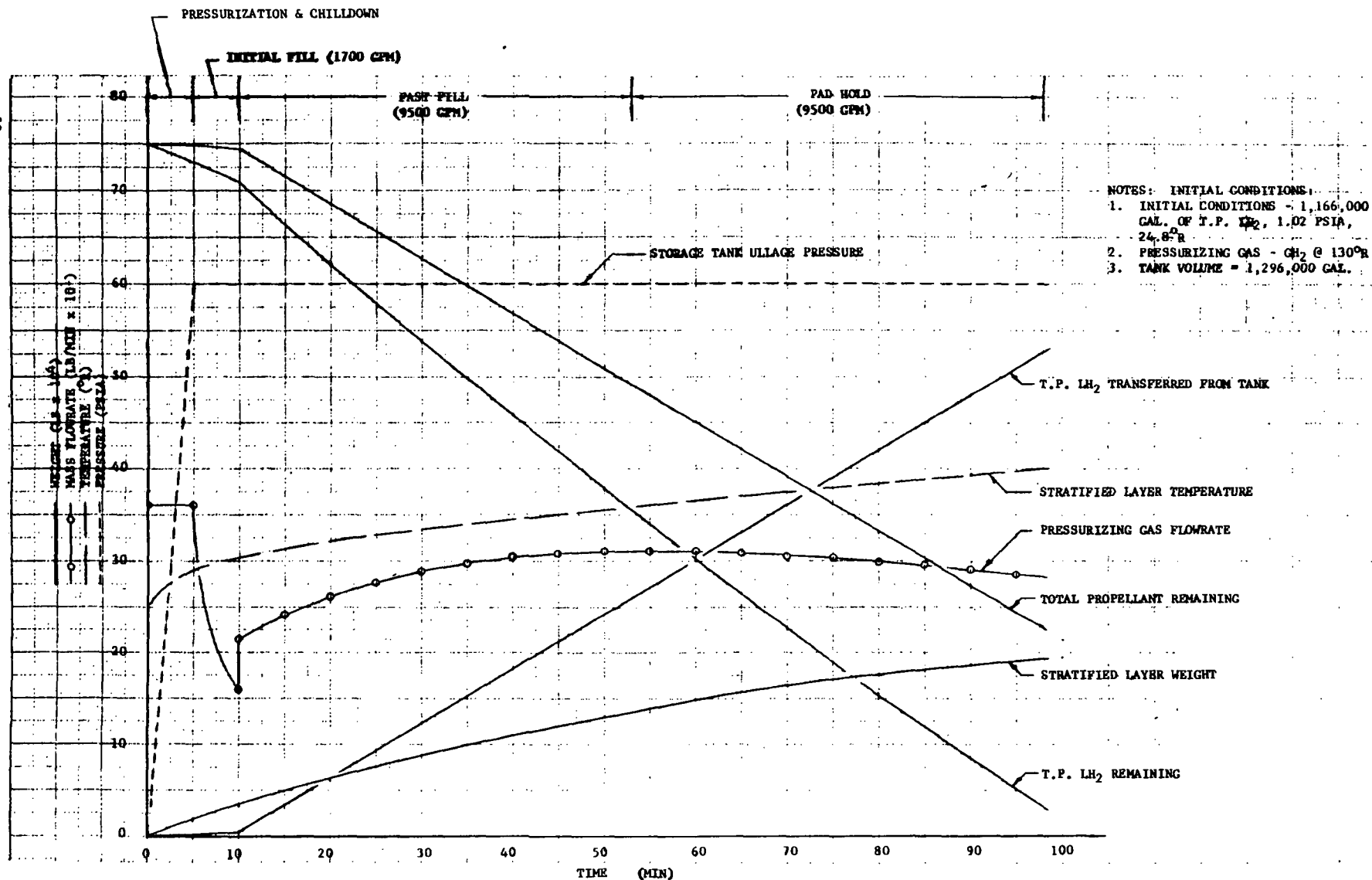


FIGURE II-26

SSTO/TP LH₂ PROPELLANT LOADING TIMELINE - STORAGE TANK PROFILE FOR EXISTING 8 & 10 INCH TRANSFER SYSTEM

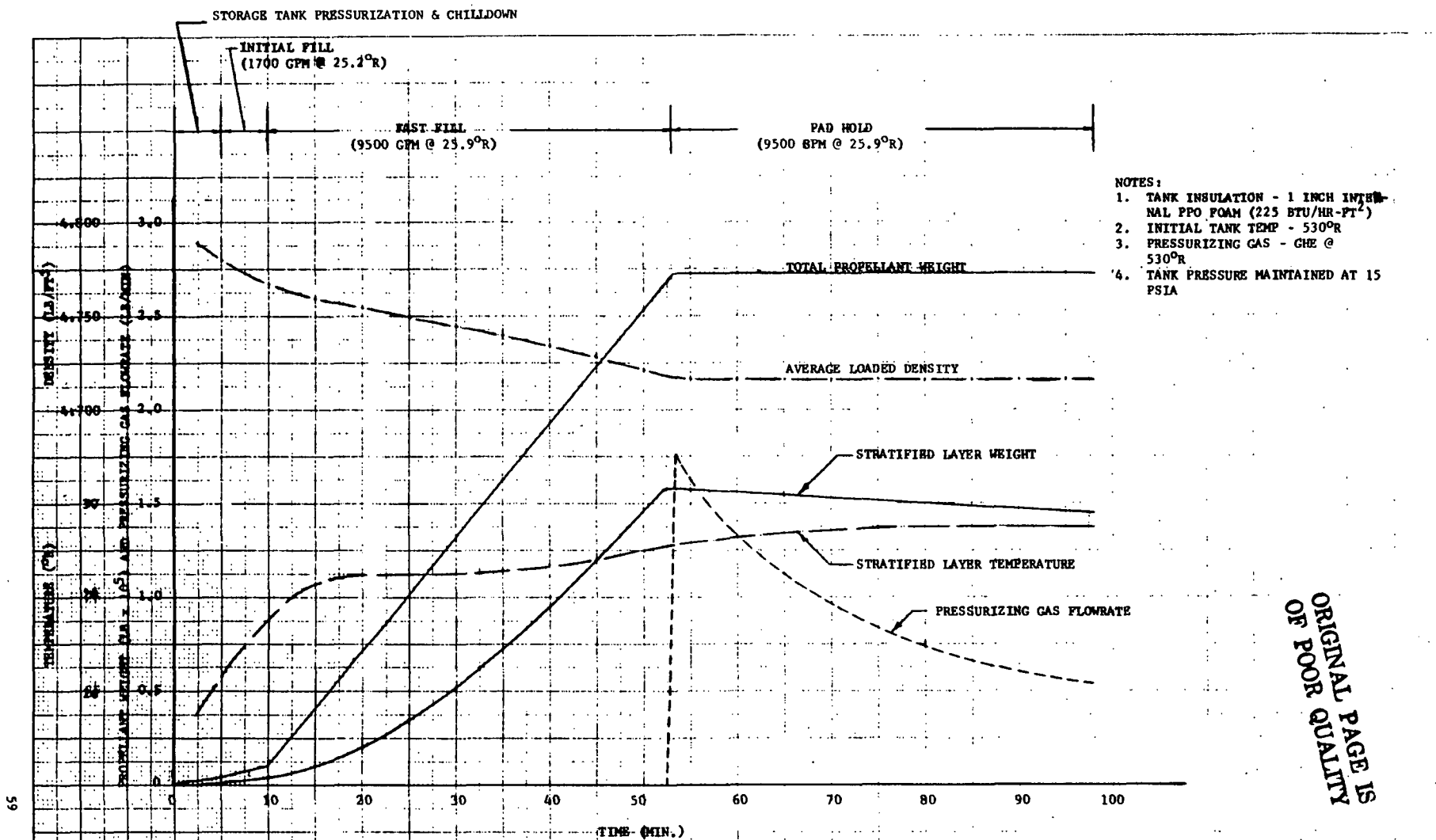


FIGURE II-27

SSTO/TP LH₂ PROPELLANT LOADING TIMELINE - VEHICLE TANK PROFILE FOR EXISTING 8 AND 10 INCH TRANSFER SYSTEM

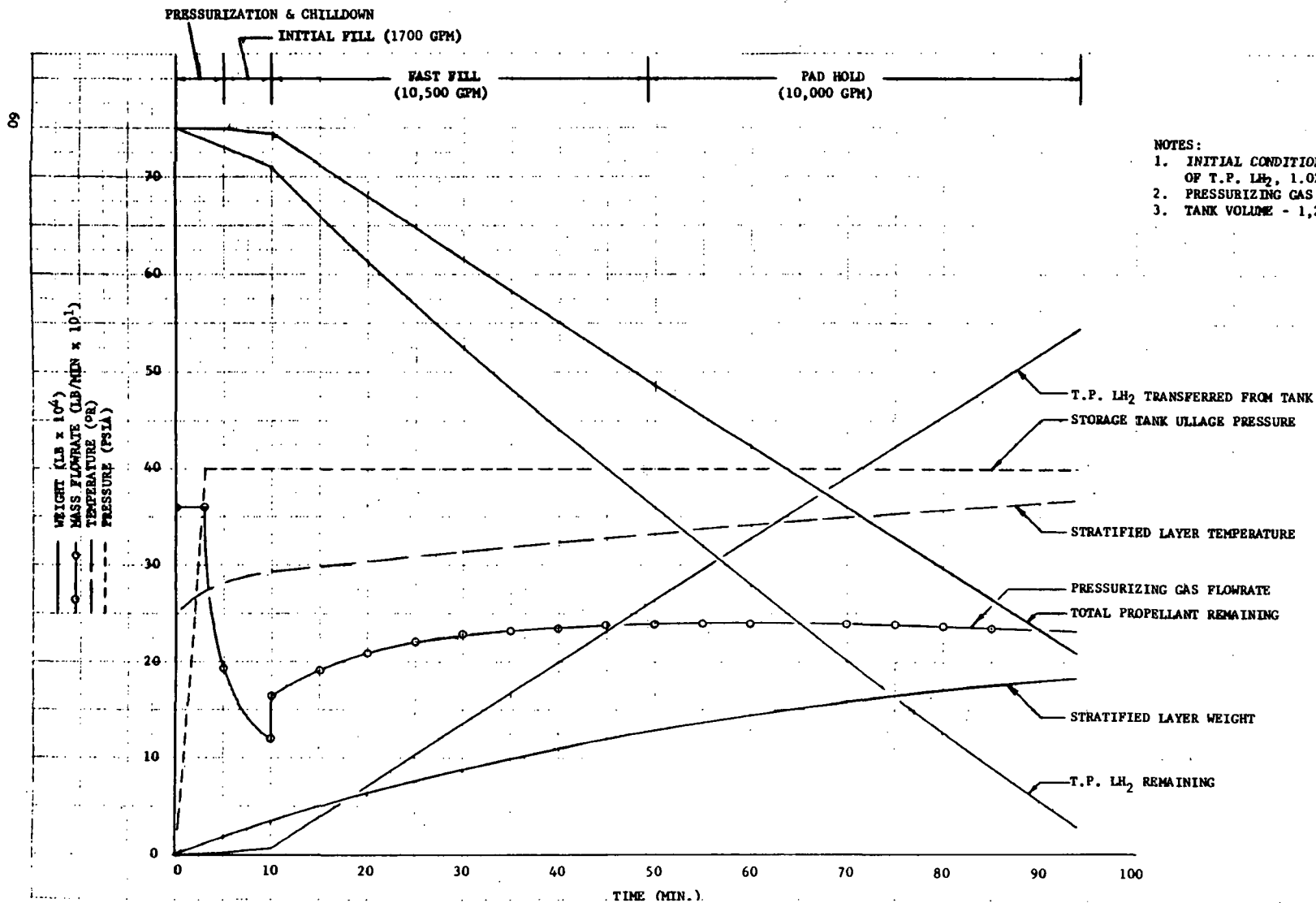
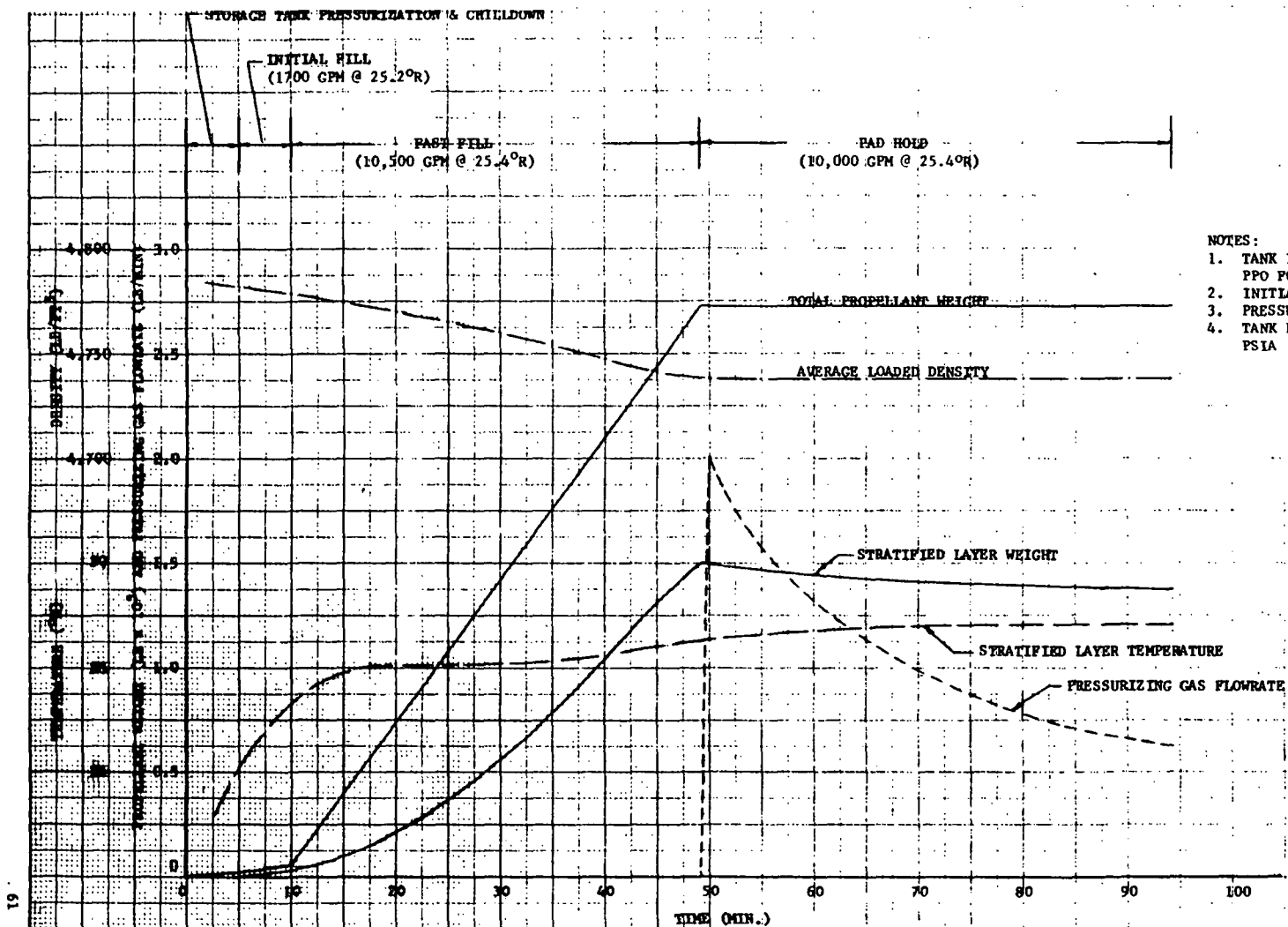


FIGURE II-28

SSTO/TP LH₂ PROPELLANT LOADING TIMELINE - STORAGE TANK PROFILE FOR 12 INCH TRANSFER SYSTEM



NOTES:

1. TANK INSULATION - 1 INCH INTERNAL PPO FOAM (225 BTU/HR-FT²)
2. INITIAL TANK TEMP. - 530°R
3. PRESSURIZING GAS - GHE @ 530°R
4. TANK PRESSURE MAINTAINED AT 15 PSIA

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FIGURE II-29

SSTO/TP LH₂ PROPELLANT LOADING TIMELINE- VEHICLE TANK PROFILE FOR 12 INCH TRANSFER SYSTEM

TP LH₂ for the entire loading sequence. Through several runs of the program, it was determined that approximately 340,200 Kg (750,000 lb) or 4414 m³ (1,166,000 gal) of TP LH₂ were required. This requires a tank volume of 4906 m³ (1,296,000 gal) if a 10% ullage is provided. In Figures II-26 and II-28, it is noted that only 113.6 m³ (30,000 gal) of TP LH₂ remain in the tanks after loading.

This significant increase (38%) in tank size for TP LH₂ compared to SLH₂ is due primarily to the absence of the solid heat of fusion. This results in higher pad hold flowrates requiring more propellant as well as a larger stratified layer in the storage tank.

Aside from the absence of the heat of fusion, a major contributor to the large increase in the stratified layer is the higher storage tank pressure required to maintain the pad hold flowrate. In the SLH₂ system, the storage tank ullage pressure was allowed to decay to 207 KPa (30 psia) to sustain the pad hold flowrates which reduces the heat transfer considerably. For the TP LH₂ systems, the pad hold flowrates are practically the same as the fast fill flowrates and the storage tank pressure cannot be reduced.

- e. Storage Tank Pressurization - The storage tank ullage pressure is shown in Figures II-26 and II-28. Since the pad hold rates for TP LH₂ are essentially the same as the fast fill rates, the tank pressure cannot be reduced after fast fill. Therefore, the tank pressure remains constant at 414 KPa (60 psia) for the 20/25 cm (8/10-inch) system and 276 KPa (40 psia) for the 30.5 cm (12-inch) system. These values are the summation of the transfer line pressure drop at the appropriate fast fill rate from Figure II-8 and the elevation head of 56 KPa (8.1 psi).

The maximum CH₂ gas flowrate required during the fast fill sequence is 141 Kg/min (310 lb/min) and 109 Kg/min (240 lb/min) for the 20/25 cm (8/10-inch) and 30.5 cm (12-inch) systems, respectively. By utilizing the primary and redundant vaporizers of the existing LH₂ system these flowrates can be obtained.

- f. Vehicle Tank Pressurization - The flowrate of 294°K (530°R) GHe required to maintain 103 KPa (15 psia) in the vehicle tank and prevent its implosion, is shown in Figures II-27 and II-29 for the 20/25 cm (8/10-inch) and 30.5 cm (12-inch) system, respectively. A maximum flowrate of 0.81 Kg (1.8

lb/min) is required for the 20/25 cm (8/10-Inch) system with a total of 26 Kg (57 lb) of helium consumed during loading and hold. For the 30.5 cm (12-Inch) system, a 0.9 Kg/min (2.0 lb/min) maximum flowrate is required and a total of 31 Kg (68 lb) of helium consumed.

- g. SSTO/TP/LH₂ Consumption Rate - The rate at which TP LH₂ is consumed or must be produced to support SSTO flights is the sum of the vehicle flight load plus the equivalent refrigeration required to make up for storage and transfer enthalpy increases. The refrigeration loss was calculated in the same manner as previously described for SLH₂ and is tabulated in Table II-9 and shown graphically in Figure II-30.

Table II-9. TP LH₂ Storage and Transfer Refrigeration Loss

Heat Source	Units	20/25 cm (8/10-Inch) System	30.5 cm (12-Inch) System
Storage tank Environmental	Kg/day (Tons/day)	590 (0.65)	590 (0.65)
Transfer line Chilldown	Kg/day (Tons/launch)	408 (0.45)	553 (0.61)
Storage tank Pressurization	Kg/launch (Tons/launch)	125,200 (138.00)	88,270 (97.30)
Transfer line friction and environmental	Kg/launch (Tons/launch)	16,700 (18.40)	8,070 (8.90)
Vehicle tank chilldown, environmental and pressurization	Kg/launch (Tons/launch)	70,760 (78.00)	69,800 (76.90)

The total consumption of TP LH₂ is shown in Figure II-31 as a function of launches per year. It is apparent that more propellant is required to offset enthalpy losses than for the vehicle load. This difference when compared to the SLH₂ systems is due primarily to the absence of the heat of fusion and the additional heating from storage tank pressurization discussed previously.

T.P.LH₂ STORAGE AND TRANSFER LOSSES
VS SSTS LAUNCH RATE

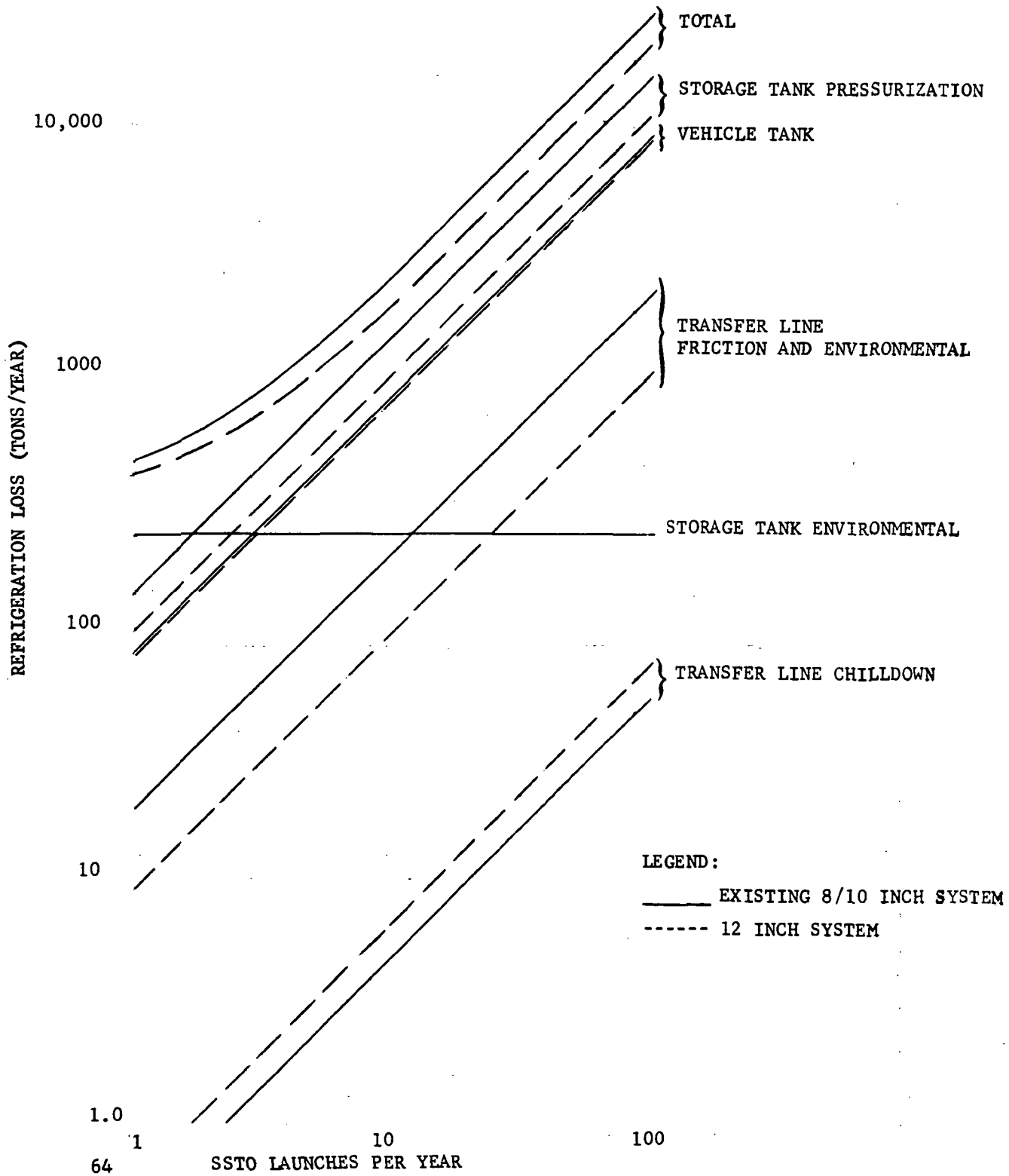


FIGURE II-30

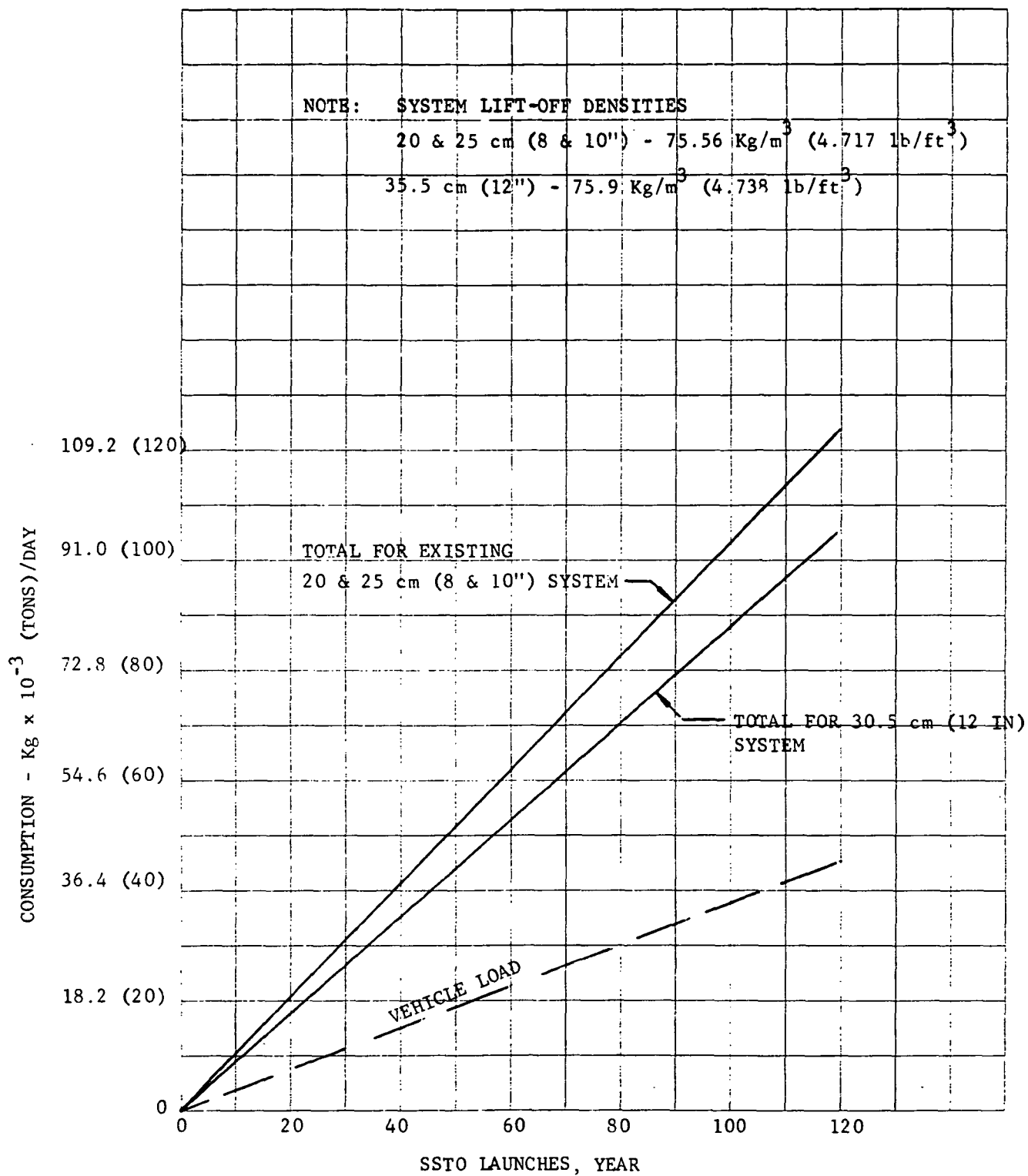


FIGURE II-31

TP LH₂ CONSUMPTION VS SSTO LAUNCH RATE

The TP LH_2 production capacity required to support the anticipated SSTO launch rate of 24 to 140 launches per year is (26 and 142 tons/day), respectively, for the existing 20/25 cm (8/10-Inch) system. For a 30.5 cm (12-inch) system this reduces to 20,000 and 112,000 Kg/day (22 and 123 tons/day) for a net savings of 3600 to 17,000 Kg/day (4 to 19 tons/day).

A 13600 Kg/day (15 ton/day) TP LH_2 production plant as prescribed by the contract Statment of Work will support 14 launches/year with the existing 20/25 cm (8/10-Inch) system and 16 launches/year with a 30.5 cm (12-Inch)

C. PRODUCTION AND COST OF DENSIFIED HYDROGEN

1. Production

Methods for the production of densified hydrogen have been presented in detail by Carney, et al (Reference 18) and are as follows:

Densified Hydrogen Production Methods

- a. Vacuum Pumping
 - 1) Straight
 - 2) Semiflow
 - 3) Branched-flow
 - 4) Cascaded
- b. Gaseous Helium Refrigeration
 - 1) Batch Process
 - 2) Flow Process
- c. Liquid Helium Cooling
- d. GHe/GH_2 Joule-Thompson Cooling
- e. Liquid Hydrogen Compression/Expansion
- f. Venturi Cooling
- g. Gaseous Helium Injection Cooling

By the analysis presented in Appendix IV, Reference 18 of the primary candidates for large scale production are reduced to

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two vacuum pumping techniques (straight and cascaded) and the two GHe refrigeration processes. These four processes, therefore, will be applied to the case of the 50% SLH₂ and TP LH₂ fueled SSTO vehicle supported by a 13600 Kg (15-ton) per day capacity production plant.

It should be noted that recent terminology for slush production is "freeze-thaw" for the vacuum pumping process and "auger production" for the GHe refrigeration process. Freeze-thaw has been demonstrated to be the most efficient vacuum pumping technique (Reference 19) and could apply to both straight and cascaded vacuum pumping production. Similarly, the slush auger (Reference 20) is a device for removing solid particules from a GHe refrigerated surface and could be applied to either batch or flow production.

To determine the most economical production method, the estimated capital and operating costs for each process were determined and plotted versus plant life. This analysis considers the production of SLH₂ or TP LH₂ from NBP LH₂ and does not include the cost of NBP LH₂. To calculate capital costs the procedure described in Reference 18 was employed. The resulting costs were adjusted to 1977 dollars with Marhsal and Stevens capital equipment index. The capital costs considered are vacuum pumps, helium gas refrigerators and production tanks. Other capital items such as vacuum jacketed (V.J.) piping, phase separators, heat exchangers, instrumentation and labor were not included. The refrigeration capacity used was that required to produce 50% SLH₂ or TP LH₂ from NBP LH₂ at a rate of 13600 Kg/day (15 ton/day). It has been shown that with a 30.5 cm (12-Inch) transfer system a 13600 Kg/day (15 ton/day) SLH₂ production capability can support an SSTO schedule of 25 launches per year (launch every 15 days). Therefore, for the SLH₂ analysis a production tank capacity of 13600 Kg/day (15 ton/day) x 15 days = 204,000 Kg (225 ton) was used. For TP LH₂ a 13600 Kg/day (15 ton/day) plant will support 16 launches year for a production tank capacity of 313,000 Kg (345 ton).

The operating cost for each method was determined from the process power required as defined by Voth in Reference 20 at a power cost of 0.03 \$/KW-hr. For the vacuum pumping processes it was assumed that the H₂ gas pumped off was reclaimed as feed stock for a LH₂ plant and its reliquefaction energy cost was included.

The resulting capital and operating costs are presented in

Table II-10 and in Figure II-32 for SLH₂, and Figure II-33 for TP LH₂. From the plotted data, it can be seen that initially the vacuum pumping methods are the most economical but for a plant life greater than 13 years for SLH₂ and 15 years for TP LH₂ the flow process with GHe refrigeration becomes the most economical. Increasing the 13600 Kg/day (15 ton/day) production rate will reduce the trade-off plant life point. Gaseous helium refrigeration becomes even more attractive when considering the advantage of production at a pressure above one atmosphere inherent with the GHe refrigeration method as opposed to production at triple point pressure 7.03 KPa (1.02 psia) with the vacuum pumping methods.

Table II-10. Densified Hydrogen Production Costs

Product	Production Method	Capital Costs (1) \$ x 10 ⁶			Operating Costs (2) \$/Yr x 10 ⁵
		Vacuum Pump (s) or GHe Refrigerator	Production Tankage	Total	
50% SLH ₂	Straight Vacuum Pumping	0.73	3.94	4.67	2.49
	Cascaded Vacuum Pumping	0.72	3.21	3.93	2.56
	GHe Refrigeration (Batch Process)	3.67	3.28	6.95	1.72
	GHe Refrigeration (Flow Process)	2.92	2.87	5.79	1.24
TP LH ₂	Straight Vacuum Pumping	0.26	2.62	5.92	1.47
	Cascaded Vacuum Pumping	0.31	4.65	4.96	1.51
	GHe Refri-	2.48	5.04	7.52	1.01

geration (Batch Process)				
GHe Refri- geration (Flow Process)	1.81	4.63	6.44	0.57

NOTE: (1) Reference 18, (2) Reference 20, (3) Costs adjusted to 1977 Dollars with the Marshal and Stevens Capital Equipment Index - 523 (1977)/202 (1964) = 2.16

2. Cost of Densified Hydrogen

Liquid hydrogen for the Shuttle program is produced by Air Products and Chemicals, Inc. at the New Orleans liquefaction plant and is delivered to KSC in 3175 Kg (7000 lb) capacity roadable tankers. The sales department of Air Products was contacted by phone for information on the current price of LH_2 . It was learned that the present contract price to NASA is 3.95 \$/Kg (1.34 \$/lb) plus \$1.41 per round trip tanker mile (1268) for a delivered price to KSC of 3.53 \$/Kg (1.60 \$/lb). This amounts to 2.95 \$/Kg (2680 \$/ton) for LH_2 plus 0.561 \$/Kg (510 \$/ton) transportation charge.

To determine the cost of various qualities of densified hydrogen, the analytical technique used by Voth of the National Bureau of Standards was employed (Reference 20). This analysis includes an estimate of plant capital costs, input power costs and operation and maintenance costs per unit of product hydrogen. The analysis does not include the cost of gaseous hydrogen feed stock and costs are based on 1973 prices. To include the cost of hydrogen feed stock and adjust to 1977 dollars the cost per pound of product from the analysis was converted to a relative cost factor and multiplied by the current cost of LH_2 to NASA of 2.95 \$/kg (1.34 \$/lb). The resulting costs as a function of propellant quality for production rates of 15, 40 and 90 ton/day are presented in Table II-11 and Figure II-34.

In Figure II-34, it is shown that for a 13600 Kg/day (15 ton/day) capacity plant, the cost of NBP LH_2 is 2.95 \$/Kg (1.34 \$/lb) while the cost of 50% SL H_2 is 3.31 \$/Kg² (1.50 \$/lb). Since it costs 0.35 \$/Kg (0.16 \$/lb) to produce 50% SL H_2 from NBP LH_2 ,

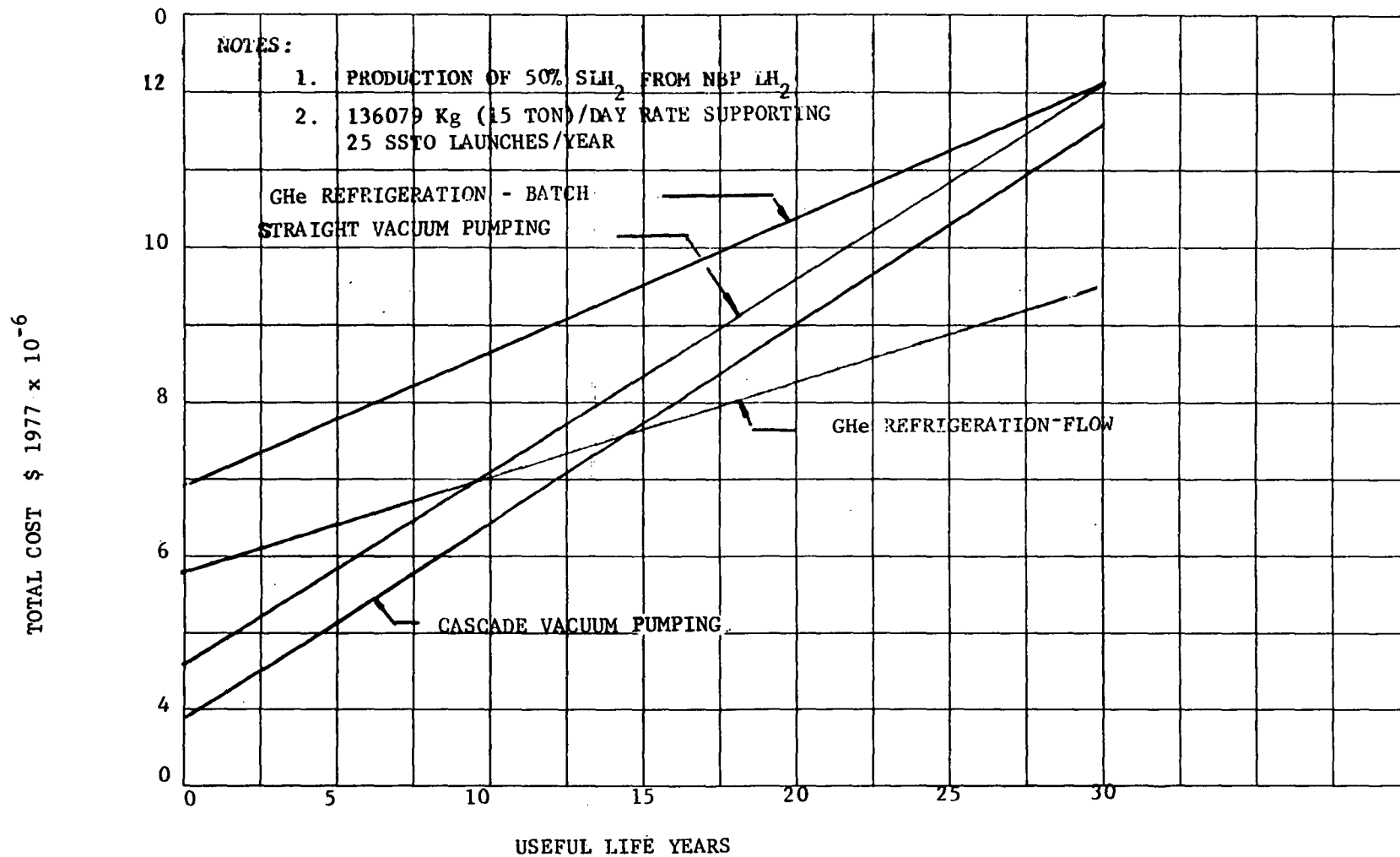
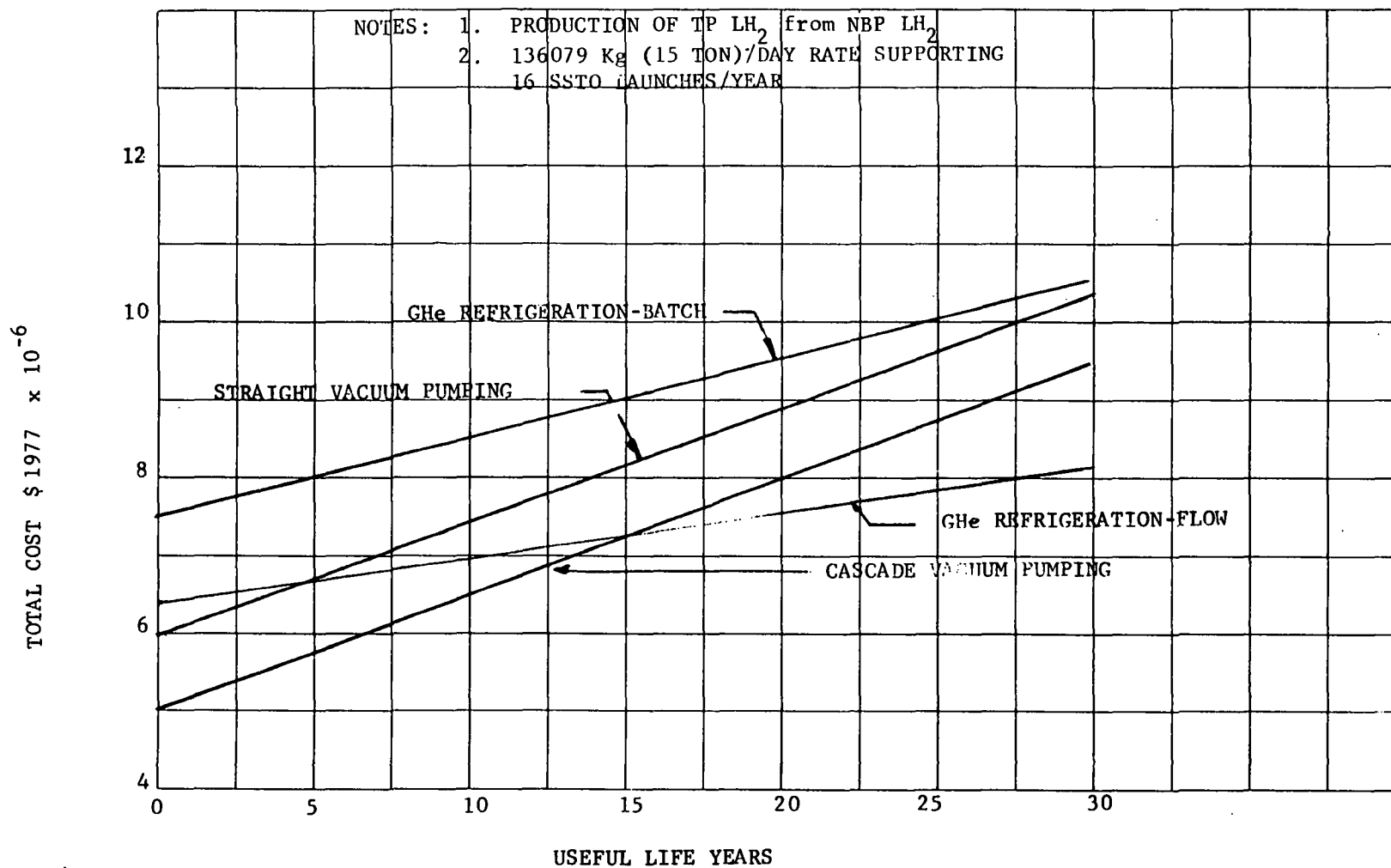


FIGURE II-32 SLUSH HYDROGEN PRODUCTION FACILITY COST VS USEFUL LIFE



71

FIGURE II-33

TP HYDROGEN PRODUCTION FACILITY COST VS USEFUL LIFE

Table II-11. Cost of Densified Hydrogen

Plant Production Rate Kg (tons/day)	Product	Relative Cost (\$/lb) (Ref. 21)	Relative Cost Factor	Cost to Produce (1977 \$/lb)
13,600 (15)	NBP LH ₂	0.366 (0.166)	1.000	2.95 (1.34)
	TP LH ₂	0.395 (0.179)	1.078	3.17 (1.44)
	50% SLH ₂	0.410 (0.186)	1.120	3.31 (1.50)
	Solid H ₂	0.481 (0.218)	1.313	3.88 (1.76)
36,300 (40)	NBP LH ₂	0.280 (0.127)	1.000	2.27 (1.03)
	TP LH ₂	0.304 (0.138)	1.087	2.47 (1.12)
	50% SLH ₂	0.313 (0.142)	1.118	2.54 (1.15)
	Solid H ₂	0.362 (0.164)	1.291	2.93 (1.33)
81,600 (90)	NBP LH ₂	0.243 (0.110)	1.000	1.96 (0.89)
	TP LH ₂	0.260 (0.118)	1.073	2.09 (0.95)
	50% SLH ₂	0.269 (0.122)	1.109	2.18 (0.99)
	Solid H ₂	0.304 (0.138)	1.254	2.47 (1.12)

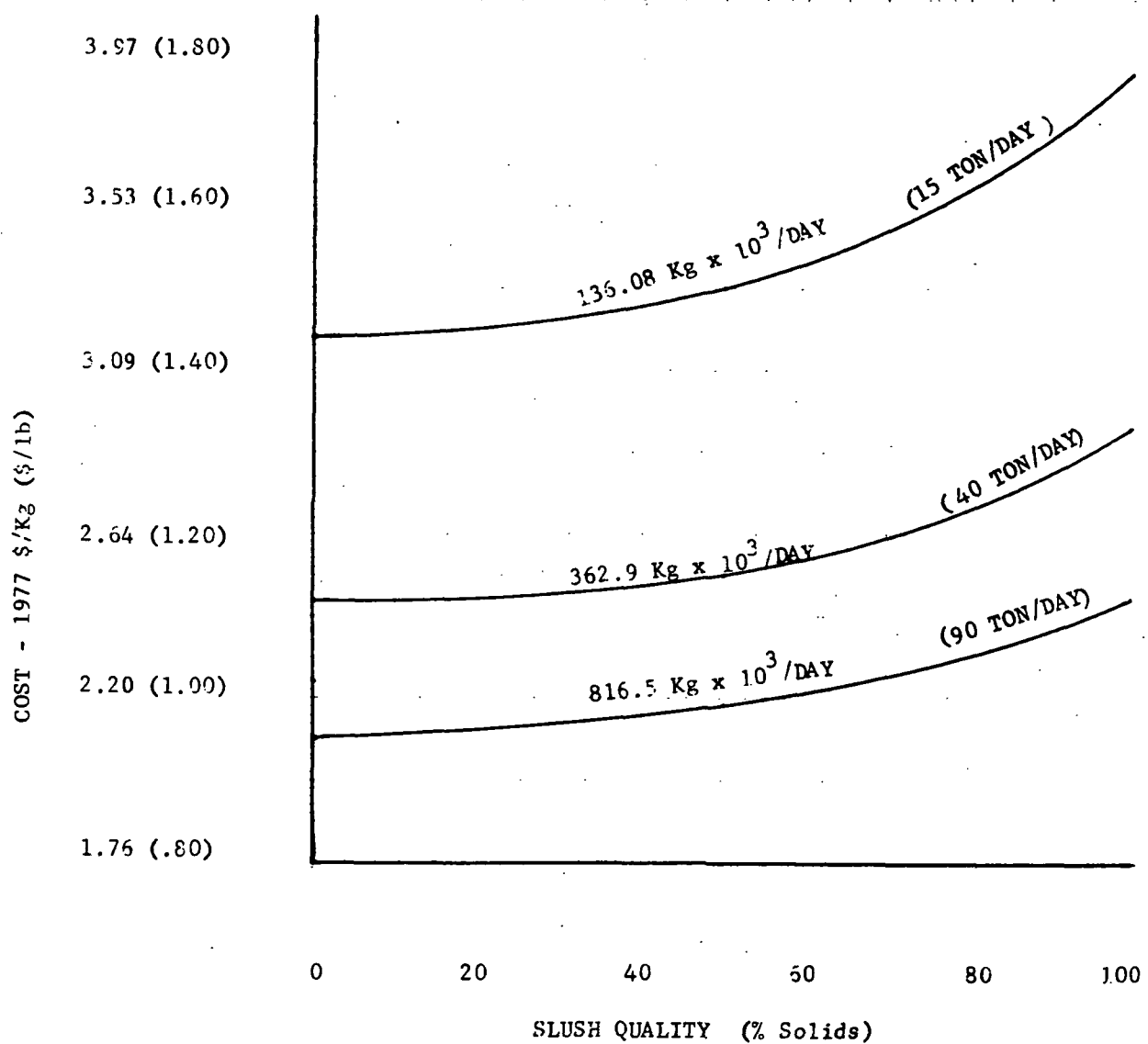


FIGURE II-34 COST OF DENSIFIED LIQUID HYDROGEN VS QUALITY FOR VARIOUS PRODUCTION RATES

then the cost per percent change in quality from 50% is $0.35 \times 2/100 = 0.007$ \$/Kg-% ($0.16 \times 2/100 = 0.0032$ \$/lb-% or 6.40 \$/ton-%). For a production capacity of 82000 Kg/day (90 tons/day), the price is reduced to 2.18 \$/Kg (0.99 \$/lb) for 50% SLH₂ and the cost per percent change in quality from 50% is 0.044 \$/Kg (4.0 \$/ton).

Similarly, in Figure II-34, it is shown that for a 13600 Kg/day (15 ton/day) capacity plant the cost of TP LH₂ is 3.17 \$/Kg (1.44 \$/lb) or 0.22 \$/Kg (0.10 \$/lb) to produce from NBP LH₂. Since the difference in temperature between TP and NBP LH₂ is 6.43°K (11.58°R) then the cost per degree rise in temperature is $0.22/6.43 = 0.03420$ \$/Kg-°K ($0.10/11.58 = 0.00863$ \$/lb-°R or 17.27 \$/ton-°R). For a 81600 Kg/day (90 ton/day) plant, the TP LH₂ costs are reduced to 2.09 \$/Kg (0.95 \$/lb) and 0.00198 \$/Kg-°K (10.36 \$/ton-°R).

3. Location of Hydrogen Densification Plant

There are three rational options for the location of a densification plant to support SSTO launches from KSC. These are:

- Option A - Locate densification plant at an existing LH₂ plant and transport densified propellant to KSC.
- Option B - Locate densification plant at KSC and transport LH₂ from existing plant.
- Option C - Locate densification plant and new LH₂ plant at KSC.

Consider Options A and B:

The methods presently being studied for the production of large quantities of TP or slush hydrogen cool NBP LH₂ by one means or another. It can be assumed, therefore, that the densification plant will be a separate facility whose feed stock is NBP LH₂ and its design and cost will be basically the same whether located next to an LH₂ plant or not.

It has been previously shown that significant losses in propellant quality or density occur during vehicle loading operations which result in large quantities of low density propellant that must be upgraded. It will be shown later (Para. V. B.) that ground transportation of densified hydrogen results in significant

losses.

The cost of SLH_2 and $TP LH_2$ delivered in roadable tankers to KSC from New Orleans is shown in Tables II-12 and II-13 for production plant capacities of 13,600 Kg/day (15 ton/day) and 81,600 Kg/day (90 ton/day).

For this analysis, a 24 hour delivery time from New Orleans to KSC was used and only the losses from environmental heat leak of the tanker during transit were considered. Losses associated with tanker chilldown and loading and storage tank loading are not included. Therefore, in view of the significant transportation costs and the requirement for upgrading capability at KSC, it becomes apparent that the SLH_2 plant should be located at KSC within reasonable proximity of the storage and transfer system to allow for pipeline transfer.

Consider Options B and C:

As mentioned previously, LH_2 for the Shuttle program is transported via roadable tanker from New Orleans. This plant is the closest and, according to the Air Products Sales Department, has been shown to be the only economically feasible supplier for KSC. In addition, in-house studies by Air Products have shown that roadable tankers are the safest, most economical and most reliable means for LH_2 transportation from New Orleans to KSC when compared with rail and barge shipment. Therefore, only LH_2 transportation in roadable tankers will be considered.

To determine the feasibility of constructing a new LH_2 plant at KSC, the cost of the plant was compared to the cost of transporting LH_2 from New Orleans and is presented in Figure II-35. The LH_2 plant cost as a function of capacity was obtained from data presented by Voth in Reference 21 and was converted to 1977 dollars with the Marshal and Stevens capital equipment cost index. The transportation costs were calculated for a tanker LH_2 boiloff rate of 0.5% per day assuming 24-hour transit time and the cost of LH_2 as previously defined. Again, tanker on-loading and offloading losses are not included. This analysis further assumes that feed stock is equally available at either plant location and that power costs are the same. It is understood that natural gas availability in Florida is questionable but other sources of hydrogen gas, such as electrolysis of water, may become feasible and would be subject of a separate study. In Figure II-35, it is shown that the cost of a new LH_2 plant at KSC amortized over a 15 and 25 year plant life, is an order of magnitude less than the costs incurred in transporting LH_2 from New Orleans.

Table II-12. Cost of SLH₂ Delivered to KSC from New Orleans

	Units	Production	Capacity (Delivery Rate)
		13,600 Kg/day (15 ton/day)	81,600 Kg/day (90 ton/day)
Quality Decay for 24 hr. Delivery	%/Delivery	3.8	3.8
Cost Per % Quality Loss	\$/Kg-\$(/ton-%)	0.00705 (6.40)	0.00441 (4.00)
Net Loss	\$/Kg (\$/ton)	0.0268 (24.32)	0.0168 (15.20)
Transportation Cost	\$/Kg (\$/ton)	0.561 (510)	0.561 (510)
Total Cost, New Orleans to KSC	\$/day	8015	47,200

Table II-13. Cost of TP LH₂ Delivered to KSC from New Orleans

	Units	Production	Capacity (Delivery Rate)
		13,600 Kg/day (15 ton/day)	81,600 Kg/day (90 ton/day)
Temperature Rise for 24 hr. Delivery	°K/Delivery (°R/Delivery)	0.34 (0.61)	0.34 (0.61)
Cost Per Degree Temperature Rise	\$/Kg-°K (\$/ton-°R)	0.0208 (10.53)	0.0125 (6.32)
Net Loss	\$/Kg (\$/ton)	0.0071 (6.42)	0.0042 (3.85)
Transportation Cost	\$/Kg (\$/ton)	0.561 (510)	0.561 (510)
Total Cost, New Orleans to KSC	\$/day	7750	46,200

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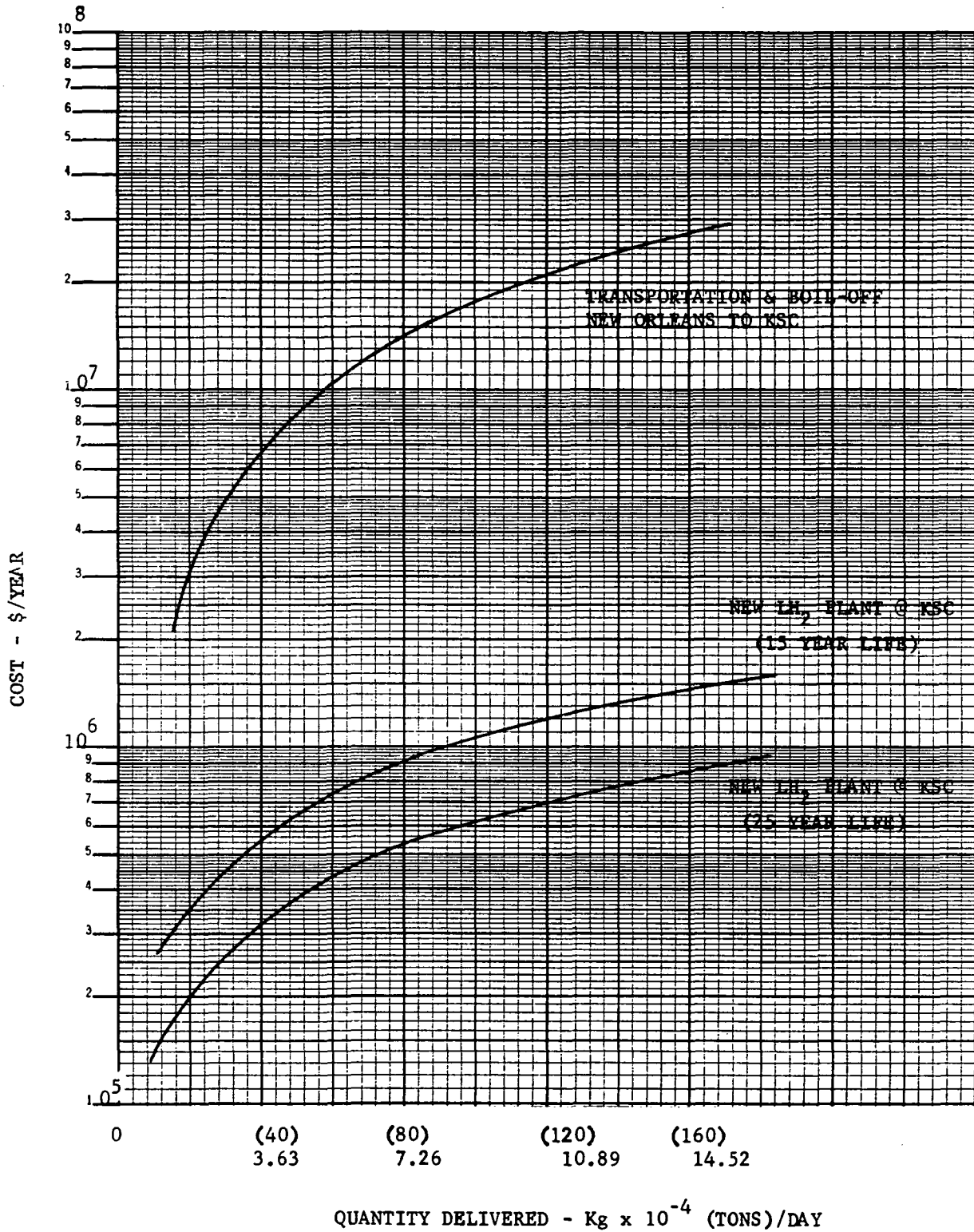


FIGURE II -35 LH₂ PLANT LOCATION FOR SSTD-COST VS QUANTITY DELIVERED

Therefore, for this study it will be assumed that both LH_2 and SLH_2 are produced at KSC.

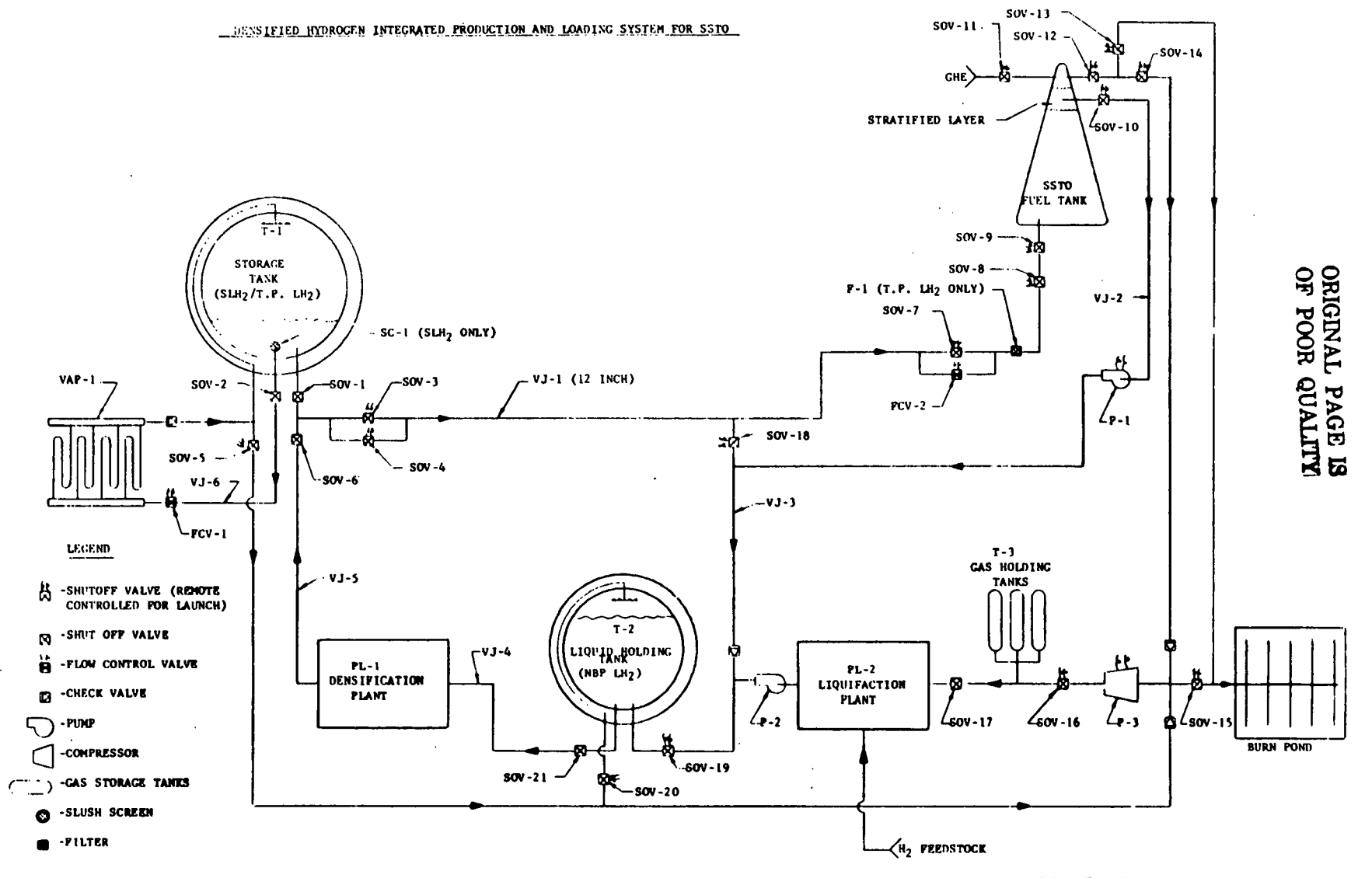
D. DENSIFIED HYDROGEN INTEGRATED PRODUCTION AND LOADING SYSTEM

The system shown schematically in Figure II-36 is the recommended integrated system for production and loading of the SSTO fuel tank with densified propellant either slush or triple point liquid hydrogen. Component identification and basic design requirements are presented in Table II-14. The system and component capacities defined therein are based on the SSTO loading sequence for a 12-inch transfer system defined previously for SLH_2 and TP LH_2 and includes a 45-minute pad hold period per launch. Production rates and storage capacities are also based on a densification plant capacity of 15 tons per day as defined by the contract statement of work. This capacity will support twenty-five (25) 50% SLH_2 fueled SSTO launches/year and sixteen (16) TP LH_2 fueled SSTO launches/year.

It has been shown that due to the significant transportation costs and tanker boiloff losses, the hydrogen liquefaction and densification plants should be located near the launch site so that filling of the storage tank via pipeline is feasible. Also, since the nature of densified propellants requires a system to upgrade and/or maintain a predefined on-board density, the loading system must be capable of transfer and storage of low density propellant from the flight tank. Therefore, the system recommended for loading the SSTO with densified hydrogen consists of the following:

- Storage tank of sufficient capacity to supply flight load and make-up for enthalpy increases due to system heating;
- Liquid holding tank of sufficient capacity to hold low density propellant transferred from vehicle during upgrading and pad hold sequences and from the storage tank at end of loading operation;
- Gas holding tank of sufficient capacity to store vaporized liquid hydrogen resulting from chilldown, pressurization and venting operations;
- Densification plant of 13600 Kg/day (15 ton/day) capacity;
- Liquefaction plant of sufficient capacity to reliquify the vaporized LH_2 and replace that consumed by the launch vehicle;

— DENSIFIED HYDROGEN INTEGRATED PRODUCTION AND LOADING SYSTEM FOR SSTS —



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FIGURE II-36

DENSIFIED HYDROGEN INTEGRATED PRODUCTION AND LOADING SYSTEM FOR SSTD
COMPONENT IDENTIFICATION AND DESIGN REQUIREMENTS (1)

Component Designation	Component Name	Capacity/Flowrate		Other Design Requirements/Remarks
		SLH ₂ System	TPLH ₂ System	
F-1	Transfer Line Filter	Not Applicable	0-10,500 gpm	1. 40-70 micron element (Note 2) 2. 5 psi (max) pressure drop at 10,500 gpm
FCV-1	Vaporizer Flow Control Valve	0-360 lb/min	0-360 lb/min	Regulates storage tank ullage pressure by controlling vaporizer flowrate (Note 2)
FCV-2	Pad Hold Flow Control Valve	0-3000 gpm	0-10,000 gpm	Regulates pad hold flowrate to maintain average loaded density
P-1	Upgrading and Hold Pump	For Upgrading: 0-11,000 gpm For Pad Hold: 0-3000 gpm	For Pad Hold Only: 0-10,000 gpm	Increasing upgrading time can reduce pump capacity considerably
P-2	Liquefaction Plant Pump	0-22 gpm	0-15 gpm	
P-3	Vent Gas Compressor	Vehicle Venting: 0-30 lb/min Storage Tank Venting: 0-95 lb/min for 1 hr	Vehicle Venting: 0-30 lb/min Storage Tank Venting: 0-160 lb/min for 1 hr	
PL-1	Densification Plant	15 ton/day	15 ton/day	1. Flow process gaseous helium refrigeration system 2. Must be capable of processing liquid hydrogen varying in temperature from 25 to 36°R
PL-2	Liquefaction Plant	9.4 ton/day	6.3 ton/day	Current technology plant
SC-1	Storage Tank Slush Screen	0-360 lb/min	Not Applicable	30 mesh screen to retain solid particles in tank
SOV-1	Storage Tank Shutoff Valve (Transfer Line)	0-11,000 gpm	0-10,500 gpm	(Note 2)
SOV-2	Storage Tank Shutoff Valve (Vaporizer)	0-560 gpm	0-560 gpm	(Note 2)
SOV-3	Transfer Line Valve	0-11,000 gpm	0-10,500 gpm	(Note 2)
SOV-4	Childdown and Initial Fill Valve	0-1560 gpm	0-1700 gpm	(Note 2)
SOV-5	Storage Tank Vent Valve	0-95 lb/min for 1 hr.	0-160 lb/min for 1 hr	1. Gas Temp = 90°R 2. Flowrate will be considerably higher for emergency venting capability
SOV-6	Storage Tank Fill Valve	30.2 gpm	32.4 gpm	(Note 2)
SOV-7	Fast Fill and Upgrading Valve	0-11,000 gpm	0-10,500 gpm	Two position valve if upgrading flowrate is not same as fast fill rate (Note 2)
SOV-8	Debris Valve	0-11,000 gpm	0-10,500 gpm	Ground side of disconnect (Note 2)
SOV-9	Vehicle Fill and Drain Valve	0-11,000 gpm	0-10,500 gpm	Vehicle side of disconnect (Note 2)
SOV-10	Upgrading and Hold Valve	0-11,000 gpm	0-10,500 gpm	
SOV-11	Vehicle Ullage Pressure Vent Valve	0-1.7 lb/min	0-2.0 lb/min	GHE @ 530°R
SOV-12	Vehicle Vent Valve	0-30 lb/min	0-30 lb/min	GH ₂ @ 200°R (Note 2)
SOV-13	Contaminated Vent Gas Valve	0-30 lb/min	0-30 lb/min	GH ₂ and GHE @ 200°R
SOV-14	Vent Line Isolation Valve	0-30 lb/min	0-30 lb/min	GH ₂ @ 200°R
SOV-15	Burn Pond Valve	----	----	Size dependent on emergency venting time requirement
SOV-16	Vent Gas Compressor Discharge Valve	0-95 lb/min	0-160 lb/min	
SOV-17	Liquefaction Plant Shutoff Valve	----	----	
SOV-18	Storage Tank Drain Valve	0-3000 gpm	0-10,000 gpm	Transfers residual liquid from storage to holding tank after launch
SOV-19	Holding Tank Inlet Valve	0-3000 gpm	0-10,000 gpm	
SOV-20	Holding Tank Vent Valve	----	----	
SOV-21	Holding Tank Outlet Valve	0-35 gpm	0-35 gpm	
T-1	Storage Tank	650,000 gal (min)	1,150,000 gal (min)	1. Vacuum jacketed sphere (Note 2) 2. Maximum heat leak = 0.080 BTU/ft ² -hr 3. Internal working pressure = 0-45psia 4. Minimum tank volumes with 10% ullage: SLH ₂ system = 713,000 gal T.P.LH ₂ system = 1,265,000 gal
T-2	Liquid Holding Tank	300,000 gal (min)	850,000 gal (min)	1. Vacuum jacketed sphere or multiple cylinders 2. Maximum heat leak = 0.080 BTU/ft ² -hr
T-3	Gas Holding Tank	8,000 lb (min)	12,000 lb (min)	Provides for storage of gas during loading and storage tank venting after loading
VAP-1	Storage Tank Vaporizer	360 lb/min CH ₂ @ 130°R	360 lb/min CH ₂ @ 130°R	Similar design to existing LC39 ambient temperature, gravity feed heat exchanger except twice the flow capacity
VJ-1	Transfer Line	0-11,000 gpm	0-10,500 gpm	1. Multilayer insulated (MDI) vacuum jacketed line (Note 2) 2. Inner line - 12 inch inner pipe 3. Average heat leak - 12 BTU/hr-ft (max)
VJ-2	Upgrading and Hold Line	0-11,000 gpm	0-10,500 gpm	Same design as VJ-1
VJ-3	Storage Tank Drain Line	0-3000 gpm	0-10,000 gpm	Same design as VJ-1
VJ-4	Holding Tank Transfer Line	0-35 gpm	0-35 gpm	Same Design as VJ-1
VJ-5	Densification Plant Transfer Line	0-30.2 gpm	0-32.4 gpm	Same Design as VJ-1
VJ-6	Vaporizer Supply Line	560 gpm	560 gpm	Same Design as VJ-1

Notes: (1) Design requirement for 12 inch transfer system with 45 minute pad hold capability and production capacity of 15 tons/day supporting 25 SLH₂ and 16 T.P.LH₂ fueled SSTD flights per year.
(2) Similar to existing ESC LC39 configuration.

TABLE II-14

DENSIFIED HYDROGEN INTEGRATED PRODUCTION AND
LOADING SYSTEM FOR SSTD - COMPONENT IDENTIFICATION
AND DESIGN REQUIREMENTS (1)

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- Piping, control valves and transfer devices.

The system, as defined, provides for density management in the vehicle tank during upgrading and pad hold by withdrawing low density propellant from the top (stratified layer) of the tank and supplying, at the same rate, high density propellant into the bottom. For the SLH₂ system with a 2.5 cm (1-Inch) PPO foam-insulated flight tank, a flowrate of 11.3 m³/min (2990 gpm), 59% SLH₂ is required to maintain 50% SLH₂ in the flight tank during steady-state pad hold conditions. This equates to a mass flowrate of 1.35×10^6 Kg/day (1485 tons/day) which clearly shows that real time refrigeration is not feasible and that the storage tank must provide the upgrading and pad hold propellant.

A method for onboard production and density maintenance, that of refrigeration by gaseous helium bubbling, was investigated briefly. By the method of analysis described in Reference 22, it was determined that a 2.5 cm (1-Inch) PPO foam-insulated SSTO loaded with NBP LH₂ would require 5.44×10^6 Kg/day (6000 ton/day) of 11°K (20°R) gaseous helium to maintain 50% SLH₂ during pad hold.

In Figure II-36, no device is shown for the transfer of liquid from the liquid holding tank through the densification plant into the storage tank. It has been shown that considerable enthalpy increase results from transfer by either pump or pressurization discharge. In order to begin a loading operation with the highest possible density, therefore, it is suggested that gravity transfer be considered. Gravity flow is feasible since the flowrates involved are only 0.11-0.13 m³/min (30-35 gpm). If the problems associated with locating the densification plant and liquid holding tank above the 18.3 m (60-ft) storage tank can be overcome, enthalpy increases can be minimized.

Since the system is closed loop with all vented GH₂ being reclaimed, the burn pond shown in Figure II-36 will not normally be used. If emergency venting of a large volume of GH₂ is required, however, it would be routed to the burn pond for safe disposal. Also, the system as shown provides for the venting of GH₂ contaminated with GHe vehicle tank venting is required after GHe pressurization is initiated. This contaminated gas would be routed to the burn pond.

1. System Component Description

- a. Storage Tank - The storage tank is a vacuum jacketed sphere similar in design to the existing LC 39 LH₂ storage tank (Reference 6). To minimize daily enthalpy gain, a maximum allowable heat leak of 0.252 w/m^3 ($0.080 \text{ Btu/ft}^2\text{-hr}$) should be imposed which has been achieved in the existing LC 39 Pad A tank. The minimum tank capacities are 2460.5 m^3 (650,000 gal) and 4353 m^3 (1,150,000 gal) for the SLH₂ & TP systems, respectively. These values were determined from Figure II-16 and II-28 by subtracting the densified propellant remaining at launch from the total densified propellant at the beginning of the loading sequence. To provide a minimum 10% ullage, the minimum tank volumes are 2706.6 m^3 (715,000 gal) for SLH₂ and 4788.5 (1,265,000) for TP LH₂.

Also, from Figures II-15 and II-28, it is shown that the internal working pressures are 0-290 KPa (0-42 psia) for the SLH₂ system and 0-276 KPa (0-40 psia) for the TP LH₂ system.

A unique requirement for the SLH₂ system is a 30-mesh screen over the vaporizer supply line outlet to conserve the solid particles by allowing only liquid to flow to the vaporizer.

- b. Liquid Hydrogen Holding Tank - For the baseline configuration this is a spherical vacuum jacketed tank of similar design and heat leak as the storage tank. The minimum tank capacities are 1135.6 m^3 (300,000 gal) for the SLH₂ system and 3217.6 m^3 (850,000 gal) for the TP LH₂ system. These capacities are the summation of the propellant transferred during the upgrading and hold sequences and the residual propellant in the storage tank and lines after loading. Consideration should be given to the use of multiple cylindrical tanks instead of a single sphere as it may be advantageous to separate the recycled liquid of different temperatures for redensification. Also, there may be economic advantages of cylindrical tanks as compared to spheres.
- c. Gas Holding Tank - A high pressure gas tube bank provides for the storage of vented gaseous hydrogen during the loading operations shown in Table II-15. The volumetric capacity of the tube bank is determined by the storage pressure which will be determined from the discharge pressure of the vent gas compressor.

Table II-15. Gaseous Hydrogen Vented During Loading

Loading Operation	Gas Vented Kg (lb)	
	SLH ₂ System	TP LH ₂ System
Transfer Line Chillydown	558 (1230)	558 (1230)
Vehicle Venting During Loading	395 (870)	426 (940)
Storage Tank Venting After Loading	2540 (5600)	4400 (9700)
Total	3493 (7700)	5384 (11870)

- d. Vent Gas Compressor - A compressor in the vent system pressurizes hydrogen gas for temporary storage until it can be reliquified. The minimum compressor capacity of 13.6 Kg/min (30 lb/min) is required to reclaim the vehicle vent gas. The maximum compressor capacity depends on the time allowed for venting the storage tank gas after loading. To vent the tank in one hour, the required compressor capacities are 43 Kg/min (95 lb/min) and 72.6 Kg/min (160 lb/min) for the SLH₂ and TP LH₂ systems, respectively.
- e. Storage Tank Vaporizer - The storage tank vaporizer is an ambient temperature heat exchanger which supplies pressurized gas to the storage tank by vaporizing TP LH₂. In order to pressurize the fully-loaded storage tank to operating pressure in 5 minutes with 72.2°K (130°R) GH₂, a flowrate of 163 Kg/min (360 lb/min) is required for both SLH₂ and TP LH₂ systems (see Figures II-16 and II-28). The maximum vaporizer flowrate during loading is 102 Kg/min (225 lb/min) and 113 Kg/min (250 lb/min) for the SLH₂ and TP LH₂ systems, respectively. The vaporizer capacity could be reduced accordingly if longer initial pressurization times are allowed.
- A design similar to the 81.6 Kg/min (180 lb/min), 72.2°K (130°R) GH₂ vaporizer presently in service at LC 39 is recommended.
- f. Upgrading and Hold Pump - Circulation of propellant from the fully-loaded vehicle tank is provided by means of the upgrading and hold pump. In the case of TP LH₂ it has been shown that within practical flowrate limits circulation can only maintain, but not increase, the loaded density. To maintain

the average loaded density during steady-state hold conditions, pump flowrates of 11.36 m³/min (3000 gpm) and 37.85 m³/min (10,000 gpm) are required for the SLH₂ and TP LH₂ systems, respectively.

For SLH₂ system, the upgrading flowrate was assumed to be equal to the fast fill rate of 41.6 m³/min (11,000 gpm) in order to determine the minimum time required to achieve 50% SLH₂ density. In practice, the upgrading can be accomplished during the pad hold period thus requiring a smaller upgrading system.

- g. Liquefaction Plant Pump - Transfer of NBP LH₂ from the liquefaction plant to the densification plant is provided by means of a .08 m³/m (0-22 gpm) pump for the SLH₂ system and .06 m³/m (0-15 gpm) pump for the TP LH₂ system.
- h. Other System Components - All other system components such as shutoff valves, flow control valves, check valves, filters, transfer lines and vent lines are of similar design to the existing KSC LC 39 and LH₂ system components (Reference 6).

2. System Operating Procedure

Loading of the SSTO for launch with SLH₂ or TP LH₂ is to be implemented within the timeline previously defined and by the operating procedure which follows.

Prior to beginning the loading operation, all valves shown in the loading and production system schematic (Figure II-36) are closed except for storage tank valves SOV-1 and SOV-2 and holding tank valves SOV-20 and SOV-21. The storage tank (T-1) is fully loaded and the liquid and gas holding tanks (T-2, T-3) are empty. The liquefaction and densification plants, due to their proximity to the launch site, are in an unmanned, standby mode.

- a. Storage Tank Pressurization and Chillydown - The transfer operation is initiated by opening the following transfer and vent line valves:

- Chillydown Valve (SOV-4)
- Fast Fill and Upgrading Valve (SOV-7)
- Debris Valve (SOV-8)
- Vehicle Fill and Drain Valve (SOV-9)
- Vehicle Vent Valve (SOV-9)
- Vent Line Isolation Valve (SOV-14)
- Vent Gas Compressor Discharge Valve (SOV-16)

Simultaneously, the Vaporizer Flow Control Valve (FCV-1) opens and the storage tank begins to pressurize. When the tank operating pressure is reached, FCV-1 automatically throttles to the flowrate required to maintain the tank pressure during outflow.

As liquid vaporizes in the transfer line and vehicle tank and is exhausted into the vent line, the Vent Gas Compressor (P-3) is turned on to transfer gas into the Gas Holding Tanks (T-3) while maintaining a minimal backpressure in the vehicle vent line.

During this period as the storage tank pressure increases, the flowrate increases to the initial fill rate which can be limited by an orifice in the chilldown line.

- b. Initial Fill - The flowrate is maintained constant through the Chilldown Valve until the vehicle tank is loaded to 2% of flight volume as measured by time or liquid level sensors.
- c. Fast Fill - At the 2% load signal, the Transfer Line Valve (SOV-3) opens and the flowrate increases to the fast fill rate. Simultaneously, the Vent Gas Compressor speed is increased to transfer and store the additional vent gas.
- d. Upgrading - As the flight load approaches 100% the Upgrading and Hold Valve (SOV-10) and Holding Tank Valves (SOV-19 and SOV-20) open and the Upgrading and Hold Pump (P-1) is turned on. The inlet to line VJ-2 is located in the vehicle tank whereby warm stratified liquid is withdrawn from the top of the tank and is transferred via pump P-1 into tank T-2. As liquid enters tank T-2 the displaced H_2 gas is vented through valve SOV-20 and compressor P-3 into tank T-3.

Simultaneously, the Vehicle Vent Valve (SOV-12) closes and the Vehicle Ullage Pressurant Valve (SOV-11) opens to maintain positive tank pressure.

- e. Pad Hold - When the required average loaded density is achieved, valve SOV-7 closes and the Pad Hold Flow Control Valve (FCV-2) throttles open while the speed of pump (P-1) is reduced. The flowrate through FCV-2 is controlled to maintain constant average density until launch. For the SLH_2 system, the Storage Tank Vent Valve (SOV-5) opens and storage tank pressure begins to decay.

- f. Post Launch - At liftoff, valves SOV-8, SOV-9, SOV-10, SOV-11 and SOV-12 are closed and the storage tank drain valve (SOV-18) opens. The remaining propellant from the storage tank, which is of low quality due to pressurization heating, is transferred into the liquid holding tank for redensification. SOV-16 closes, SOV-17 opens and gas from the Gas Holding Tanks is transferred to the Liquefaction Plant. Residual propellant in lines VJ-1 and VJ-2 are flushed by GHe purge (not shown). Finally, the transfer system is inerted with GHe and secured while the liquefaction and densification plants are brought on line to refill the storage tank.
- g. Burn Pond - For emergency venting of the storage or vehicle tank beyond the capability of the Vent Gas Compressor, SOV-15 is opened and the gas is routed to the burn pond for rapid safe disposal.

If the vehicle tank venting is necessary after GHe pressurization begins (after fast fill) valve SOV-14 is closed isolating the vent system and SOV-13 is opened routing the gas to the burn pond. This precludes contamination of the recycled hydrogen gas with helium.

E. OTHER DENSIFIED HYDROGEN SYSTEM CONSIDERATIONS

1. System Safety

The major physical difference in hydrogen propellant at triple point as compared to normal boiling point is its slightly colder temperature [20.3°K versus 13.8°K (36.5°R versus 24.9°R)] and its low vapor pressure [7.03 kPa (1.02 psia)]. At these temperatures, changes due to the coefficients of thermal contraction of materials compared to NBP conditions are negligible and no deleterious thermal effects will occur.

The lower vapor pressure, however, presents the problem of air being drawn into the system causing a potentially explosive mixture. It has been shown previously that the SSTO densified hydrogen storage and transfer systems operate above atmospheric pressure and negative pressures should not normally occur. However, it is feasible that during long periods between launches the storage tank pressure could decay below atmospheric without external pressurization. Therefore, the system must be designed to isolate the stored hydrogen from the atmosphere during standby operation. The current LH₂ LC 39 system provides this capability with only minor modifications.

A vacuum of less than 7.03 KPa (1.02 psia) is maintained in the storage tank annular space thus precluding air leakage through the inner tank wall. Also, the present system (Reference 5) provides a GHe blanket pressure in the transfer and tank fill lines during standby operation which precludes O_2 from entering the tank through block valves A3401 and A3402 (see Figure II-2). The storage tank vent lines are presently purged with GN_2 and should be changed to GHe to prevent N_2 from entering through the vent block valves (A3404 and A3422) and subsequently freezing in the tank. The valve stem packing in both the Fill Line (A3402) and Transfer Line (A3401) valves are purged with GHe and the bonnet flanges are welded together thus precluding a path for O_2 leakage. The only remaining paths for oxygen leakage into the tank are the tank liquid level and ullage pressure sensing systems (A3425, A3426 and A3428), the storage tank sample valve (A3427) and the actuator shafts of vent valves A3402 and A3422 which should be enclosed with a GHe blanket pressure.

No other system safety problems peculiar to densified hydrogen were identified by this study and, in general, the same safety precautions used for liquid hydrogen are appropriate for slush and triple point hydrogen.

2. Slush Filtration

A major system problem with the use of slush hydrogen as a launch vehicle propellant is that of filtration. The current LH_2 system employs two filters (one in the transfer line and one in the storage tank fill line) whose elements are designed to remove all particles larger than 70 microns in diameter (Reference 6). Since slush particles are on the order of 1 mm diameter and larger (Reference 19), they will be restricted by the filter. Therefore, to filter by conventional means the element must be located in the vehicle engine feedline downstream of the point where solid particles have melted. Also, development of alternate filtration methods, such as electrostatic, should be pursued but show little promise due to the high flow-rates involved.

3. Slush Screens

It has been demonstrated experimentally that the solid particles in slush hydrogen can be separated from the liquid by passing the mixture through a 30-mesh screen (Reference 12). The liquid flows through the solid particules and screen and results in an increase in density on the upstream side of the screen.

This phenomenon can be used to advantage in the SLH_2 system where only liquid is needed for tank pressurization or mixing. A screen over the tank outlet duct to the storage tank vaporizer or mixing pump will conserve the solid particles and increase the tank average density.

F. DENSIFIED HYDROGEN SYSTEM CONCLUSIONS

The preceding analyses have established baseline designs for the systems required to load an SSTO vehicle with and produce the required quantities of densified liquid hydrogen. The optimum loading sequences for the vehicle utilizing either 50% SLH_2 or TP LH_2 have also been defined. In addition, the optimum vehicle LH_2 tank insulation has been determined for both types of densified LH_2 .

For the loading system, the existing 3217.6 m³ (850,000 gal) storage tank was determined to be acceptable for the slush case while a new vessel of 4542.5 m³ (1,200,000 gal) capacity is required for the triple point system. The existing KSC LC 39 tank pressurization vaporizers, when employed simultaneously, provide the required GH_2 flow to effect transfer of the densified hydrogen.

The existing 20 and 25 cm (8/10-inch) diameter multilayer insulation vacuum jacketed transfer lines at LC 39 were determined to be adequate with either slush or triple point hydrogen. However, the system studies indicated that an increase in diameter to 30.5 cm (12-Inch) while maintaining the same design improved overall system efficiency to the extent that this size line was recommended.

The optimum fast fill flowrates for loading densified hydrogen were shown to be essentially the same as presently employed for LH_2 loading of the Shuttle ET. The SSTO loading rates for a 30.5 cm (12-Inch) system are 41.64 m³/min (11,000 gpm) and 39.75 m³/min (10,500 gpm) for the SLH_2 and TP LH_2 systems, respectively, while the current Shuttle loading rate varies from 38.81 to 45.42 m³/min (10,200 to 12,000 gpm).

It was also shown that once loaded, the vehicle could be upgraded to a predefined quality in the case of SLH_2 . For the TP LH_2 system, however, upgrading cannot be accomplished within reasonable flowrate limits and the maximum achievable average density in the vehicle tank is 75.9 Kg/m³ (4.738 lb/ft³).

The flowrates required to maintain steady-state pad-hold conditions once fully loaded were evaluated and shown to be considerably higher for densified hydrogen than for NBP LH_2 . For the baseline SLH_2 system, a flowrate of $11.3 \text{ m}^3/\text{min}$ (2990 gpm) of 60% SLH_2 at the storage tank is required to maintain 50% solid fraction in the SSTO tank. Similarly, a flowrate of $37.85 \text{ m}^3/\text{min}$ (10,000 gpm) of 13.8°K (24.9°R) TP LH_2 at the storage tank is required to maintain an average density of $75.9 \text{ Kg}/\text{m}^3$ ($4.738 \text{ lb}/\text{ft}^3$) in the SSTO tank for the TP system. The Shuttle ET pad hold replenish rate is $0.38\text{--}1.14 \text{ m}^3/\text{min}$ (100-300 gpm). The magnitude of these differences in pad hold flowrates clearly shows an advantage of using SLH_2 rather than TP LH_2 . To design a system that can achieve these flowrates while maintaining 100% flight mass for launch will be one of the major technological problems encountered, especially for the triple point liquid system.

The results of the vehicle tank insulation studies performed indicated that a system consisting of 2.5 cm (1-inch) of internal PPO foam was preferred. Primary criteria for this selection were thermal efficiency, weight, cost, and operational simplicity advantages when compared with alternate methods of LH_2 propellant tank insulation. The need for internal insulation is further amplified when considering the SSTO in which the tankage is inside of the vehicle external skin.

The overall system thermal analysis identified the sources of propellant enthalpy use and showed that by far the major contributors were the heating from storage tank pressurization and vehicle tank environmental heat leak. It was further shown that to account for the enthalpy gain and vehicle load, a production capacity for a 30.5 cm (12-inch) system of 12700 to 74390 Kg/day (14 to 82 tons/day) of 60% SLH_2 is required to support the SSTO traffic model of 24 to 140 launches per year. A production capacity of 19960 to 111,600 Kg/day (22 to 123 tons/day) is required for the 30.5 cm (12-inch) TP LH_2 system. A 15 ton/day densification plant will support 25 50% SLH_2 or 16 TP LH_2 fueled SSTO launches per year.

An analysis of the methods of producing densified hydrogen was conducted. It was shown that for a plant life less than 13 years, the freeze-thaw vacuum pumping technique was economically advantageous, whereas, the GHe Refrigeration "Auger" method was cost effective for a plant life greater than 15 years. GHe Refrigeration was recommended since it allows for production at 1 atmosphere pressure. It was also shown that both the densification and liquefaction plants should be located at the launch site.

The cost of densified hydrogen as a function of quality and production rate was also determined. For a production capacity of 13,600 Kg/day (15 ton/day) the propellant costs in 1977 dollars are 3.17 and 3.37 \$/Kg (1.44 and 1.53 \$/lb) for TP and 60% SLUSH hydrogen, respectively, based on a NBP LH₂ cost of 2.95/Kg (1.34 \$/lb). Since 13,600 Kg (15 ton/day) will support 25 50% SLH₂ and 16 TP LH₂ launches per year, a savings of over \$300,000 per launch is realized with the 60% SLH₂ fueled vehicle.

These cost savings clearly show the economic advantage of using slush rather than densified liquid hydrogen. Additionally, the significantly larger storage tank requirements and the excessive upgrading and pad hold flowrate requirements of TP LH₂ make its further consideration impractical as compared to SLH₂.

III. DENSIFIED OXYGEN DISCUSSION

The triple point liquid oxygen (TP LOX) analyses were conducted with a set of baseline requirements which included an SSTO vehicle (References 1, 2 and 3) utilizing 50% SLH₂ and TP LOX at an MR of 6:1 (see Figure II-1). This vehicle with payload capability of 29,484 kg (65,000 lb) to low earth orbit has a GLOW of 1,117,246 kg (2,463,106 lb); a TP LOX capacity of 843,570 kg (1,859,755 lb) or 645.6 m³ (170,548 gal) at a density of 1306 kg/m³ (81.56 lb/ft³).

The existing Shuttle LOX loading system at KSC LC 39 (References 23, 24 and 25) is used as baseline for the ground system analysis. This system consists of 3407 m³ (900,000 gal) LOX capacity storage dewar; a LM transfer pump, a water bath vaporizer for storage tank pressurization and approximately 565m (1850 ft) of 15 and 20 cm (6- and 8-inch) diameter vacuum jacketed multilayer insulated transfer line. This system is shown schematically in Figure III-1.

A significant difference between the SSTO and Shuttle LOX system is the relative location of the oxidizer tank in the two vehicles. The LOX tank is located below the LH₂ tank in the SSTO vehicle and is above the LH₂ tank in the Shuttle External Tank (ET) resulting in a net reduction in system head pressure [36.3m (119 ft)] for SSTO.

The loading sequence for Shuttle (Reference 26) is shown in Table III-1 and provides the basis for the loading timeline analysis.

Table III-1. LOX Loading Sequence for Shuttle

Operation	Time (min)	Percent Load	m ³ /min	Transfer Rate (gpm)
Facility Chillydown	10	0	0-0.38	(0-101)
Vehicle Chillydown	15	0	0.06-0.17	(15-46)
Initial Fill	25	0-2	0.56-1.22	(151-322)
Fast Fill	113	2-98	4.15-5.29	(1095-1398)
Topping	11	98-100	0.95-1.34	(252-353)
Replenish	46	100	0.12-0.53	(31-139)

By applying the percent load and nominal transfer rates of the Shuttle loading sequence to the SSTO tank capacity the SSTO/TP LOX baseline loading sequence shown in Table III-2 was established.

LEGEND:

26

- ☒ MANUAL VALVE ☒ FLOW METER
 🔥 AUTO SHUTOFF VALVE ☒ FILTER
 🔥 FLOW CONTROL VALVE ☐ DISCONNECT
 ☒ CHECK VALVE 🌀 PUMP
 EL = ELEVATION

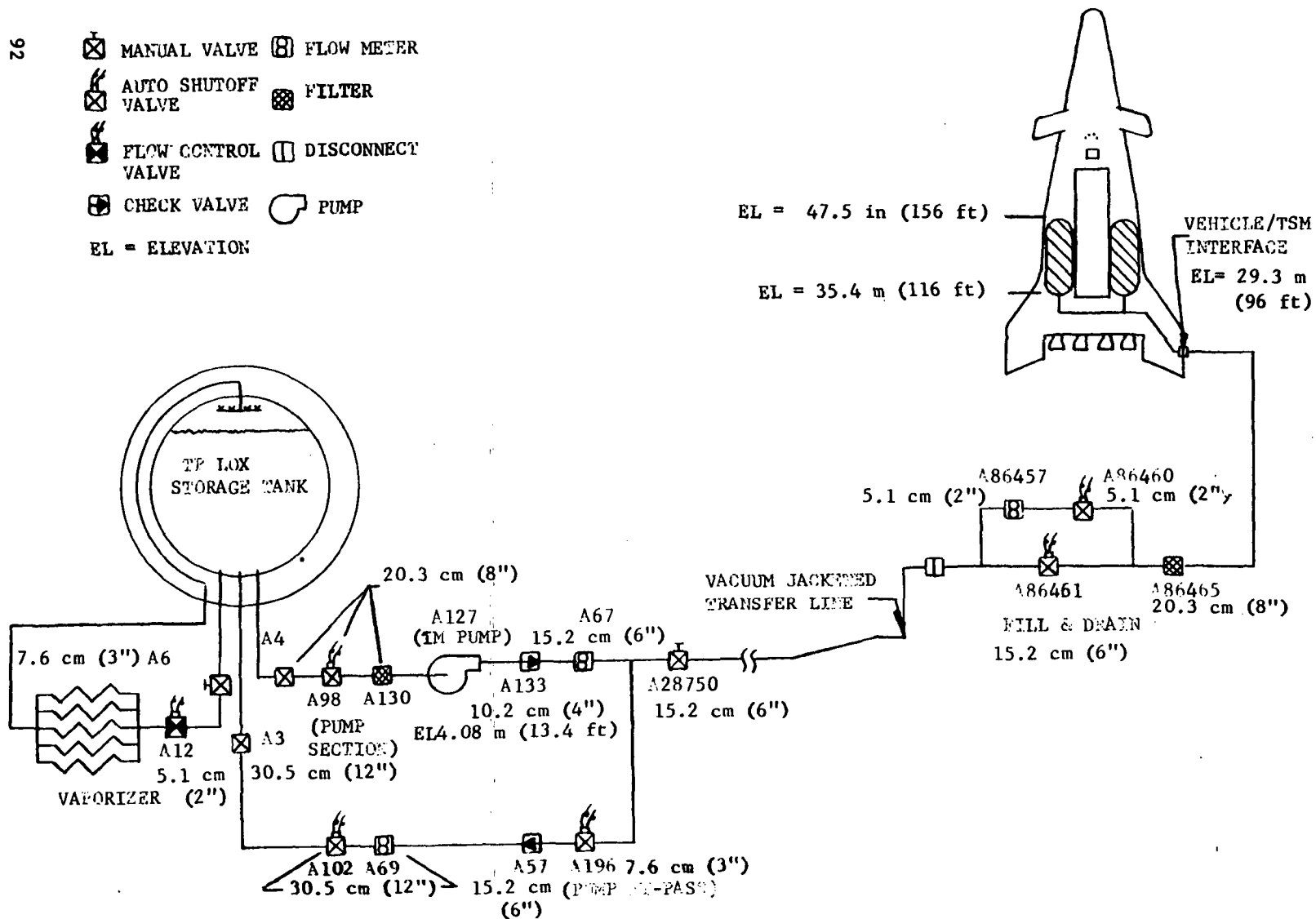


FIGURE III-1

KSC LC39 SSLO/TP LOX GASLINE LOADING SYSTEM

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Table III-2. SSTO/TP LOX Baseline Loading Sequence

Operation	Time (min)	Percent Load	Rate m ³ /min (gpm)		Quantity m ³ /min (gal)	
Chilldown	12	0	0-1.14	(0-300)	6.62	(1,750)
Initial Fill	11	0-2	1.14	(300)	12.91	(3,411)
Fast Fill	124	2-98	5.00	(1,320)	619.78	(163,728)
Topping	10	98-100	1.29	(340)	12.91	(3,411)
Replenish	46	100	0.53	(140)	24.38	(6,440)
Totals	203	100	- - -	- - -	676.60	(178,740)

A. STORAGE

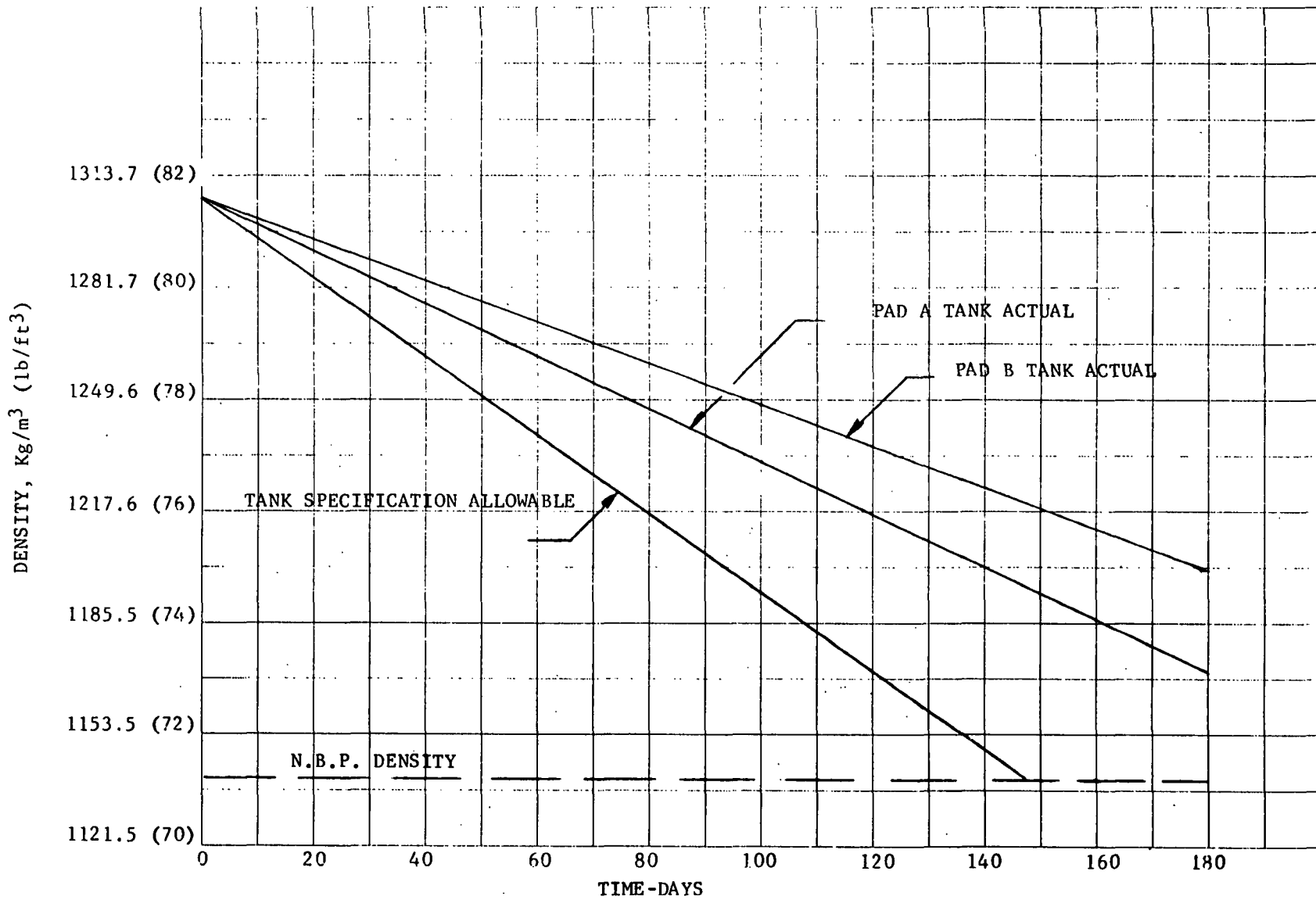
The LC 39 LOX storage tanks were designed and constructed to the requirements of Section VIII of the ASME Code for a 184 KPa (12 psig) internal working pressure (Reference 25). Each tank is insulated by a 0.9m (3-ft) annular space between inner and outer tanks filled with perlite and pressurized to slightly above ambient with nitrogen (GN₂). The tank liquid capacity is 3407 m³ (900,000 gal) of NBP LOX plus an ullage space of 10% of storage capacity resulting in a total volume of 3747.6 m³ (990,000 gal). Considering the maximum allowable capacity of the tank to be 3407 m³ (900,000 gal) at NBP LOX density [1140 kg/m³ (71.2 lb/ft³)] then the maximum allowable capacity of TP LOX at a density of 1307 kg/m³ (81.57 lb/ft³) would be 2972 m³ (785,580 gal) with a minimum ullage volume of 773.8 m³ (204,420 gal).

TP LOX cannot be stored in the existing LC 39 LOX tanks at triple point pressure [0.14 KPa (0.02 psia)]. These tanks were designed for storage of NBP LOX at a minimum pressure of one atmosphere (Reference 25). During an early Saturn V tanking test, the Pad A tank was subjected to approximately 55 KPa (8 psia) due to a system component failure and resulted in the collapsing of a portion of the inner tank (Reference 27). Therefore, in order to store TP LOX in the existing tanks, the ullage pressure must be maintained at one atm or greater. The same situation also exists in the vehicle oxidizer tank as well as the ground transport tankers used to fill the storage tank. Therefore, since TP LOX cannot be used, transported or stored in its equilibrium state, then sub-cooled LOX at triple point temperature and one atm pressure will be considered.

The primary reasons for the use of TP LOX instead of NBP LOX for the SSTO oxidizer is its increase in density [1140 to 1310 kg/m³ (71.2 to 81.6 lb/ft³)] and heat capacity [-133.53 to -193.55 kJ/kg (-57.412 to -83.216 Btu/lb)]. For sub-cooled LOX at triple point temperature and one atm pressure the density is 1310 kg/m³ (81.571 lb/ft³) and enthalpy is -193.47 kJ/kg (-83.181 Btu/lb) (Reference 28). Since the density and heat capacities are essentially the same, sub-cooled LOX at triple point temperature and 1 atm will be considered from here on and will be referred to as atmospheric triple point LOX (atm TP LOX). To provide the partial pressure necessary to maintain one atmosphere pressure in the storage tank additional gas is required. The storage tank provides for pressurization during vehicle loading by vaporizing liquid oxygen from the tank. For standby storage of TP LOX, pressurization with CO₂ is not feasible due to its low vapor pressure at the triple point temperature. Nitrogen is currently used in the LOX system as a pressurant for triple point LOX since the temperature of TP LOX [-218.8°C (-361.8°F)] is below the normal boiling point of nitrogen [-195.8°C (-320.4°F)]. The only gases with boiling points below 218.8°C (-361.8°F) are Neon [-245.9°C (-410.6°F)], hydrogen [252.7°C (-422.9°F)] and helium [-268.6°C (-451.5°F)]. Neon is not an alternative as its cost is approximately 10 to 20 times greater per cubic foot than helium depending on purity required. Hydrogen is not an inert gas and obviously cannot be used to pressurize oxygen thereby leaving only helium as a pressurizing and purge gas for TP LOX. Gaseous helium (GHe) is presently available at KSC for pressurization and purging in the liquid hydrogen systems, but its storage and transfer capacities would have to be increased substantially for use in the TP LOX systems.

As noted previously, the annular space of the storage tank is maintained at a positive blanket pressure with GN₂. To prevent nitrogen from condensing in the annular space, the outer surface of the inner sphere must be insulated or the gas changed to helium.

The specification boiloff rate for the LC 39 LOX storage tanks is 0.18% by weight of design capacity per day (Reference 25) or 5.70 m³/day (1506 gal/day). Through conversations with NASA launch operations personnel it was determined that the actual boiloff rates during Apollo and Skylab missions were approximately 3.8 and 3.0 m³/day (1000 and 8000 gal/day) for the Pad A and Pad B tanks, respectively. The resulting heat leak rates were calculated and the density decay of TP LOX in these tanks was determined and plotted in Figure III-2. The density decay rate is 0.74 kg/m³/day (0.046 lb/ft³/day) in the Pad A tank and 0.59 kg/m³/day (0.037 lb/ft³/day) in the Pad B tank. These decay rates are not excessive if the anticipated SSTO traffic model (Reference 1) is considered



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FIGURE III-2 TRIPLE POINT LOX DENSITY DECAY IN THE LC39 LOX STORAGE TANKS

which defines a minimum launch rate of one every 15 days. As was noted for the LH₂ tanks, these tanks have recently been refinished with a dark rust preventive compound that has resulted in higher boiloff rates. Since one of the emphases of this study is to minimize heat leaks, the above values will be used.

B. TRANSFER

For Shuttle, transfer of liquid oxygen is accomplished by pump transfer using a 3.8 m³/min (1000 gpm) pump previously used for replenish of the Saturn V, S-II and S-IVB stages (Reference 26). Two 38 m³/min (10,000 gpm) pumps remain in the system but are not used for Shuttle and are not considered in this study. The 3.8 m³/min (1000 gpm) pump is equipped with a variable speed clutch which can vary the pump flowrate from 3.14 m³/min (830 gpm) to 5.00 m³/min (1320 gpm). During fast fill all flow through the pump is directed to the vehicle. During the initial slow fill, topping and replenish the pump flow is reduced to 3.14 m³/min (830 gpm) which provides the necessary head pressure. The flowrate to the vehicle is adjusted by recirculating through a bypass loop into the storage tank and by changing valve positions.

By utilizing the baseline SSTO TP LOX loading sequence the system piping drawings and specifications (References 24 and 25) and the system elevation and L/D values prescribed in Reference 29, the LOX system pressure drops for varying flowrates of NBP and atm TP LOX were calculated (see Table III-3). It is significant to note that although the net GSE and vehicle pressure drops are higher for atm TP than for NBP, the transfer pump discharge pressure is less due to the relative orientation of the oxidizer tank in the two vehicles.

For the ground system from the pump to the Vehicle/Tail Service Mast (TSM) interface, the pressure drop attributable to line friction loss was calculated using the standard Darcy equation and Moody Diagram for flow in smooth pipes. Due to the uncertainty of the actual SSTO vehicle fill line configuration, the SSTO system ΔP was estimated by converting the known Shuttle ICD pressure requirement (Reference 26) to atm TP LOX density.

1. Storage Tank Pressurization

NPSH for the transfer pump is provided for in the existing system by pressurizing the storage tank to 170 KPa (10 psig) with gaseous oxygen (GO₂) via the 4.5 kg/sec (10 lb/sec) (maximum) vaporizer. To

Table III-3. LC 39 LOX Transfer System Pressure Drop vs Flow Rate for NBP LOX and Atm TP LOX

	Pressure Drop, KPa (psi)					
	Shuttle (NBP LOX)			SSTO (TP LOX)		
	Fast Fill	Topping	Replenish	Fast Fill	Topping	Replenish
Vehicle Flowrate m ³ /min (gpm)	5.00 (1320)	1.29 (340)	0.53 (140)	5.00 (1320)	1.29 (340)	0.53 (140)
Transfer Pump Flowrate m ³ /min (gpm)	5.00 (1320)	3.14 (830)	3.14 (830)	5.00 (1320)	3.14 (830)	3.14 (830)
By-pass Loop Flowrate m ³ /min (gpm)	0	1.85 (490)	2.61 (690)	0	1.85 (490)	2.61 (690)
Transfer Line Pressure Drop (Piping & Components)	498 (72.2)	41 (5.9)	254 (36.9)	666 (96.6)	55 (8.0)	293 (42.5)
GSE Heat Pressure Drop (Pump Discharge Line to Vehicle/TSM Interface = 34.1 m (112 ft))	383 (55.5)	383 (55.5)	383 (55.5)	439 (63.6)	439 (63.6)	439 (63.6)
Net GSE Pressure Drop	880 (127.7)	423 (61.4)	637 (92.4)	1105 (160.2)	494 (71.6)	732 (106.1)
Shuttle ICD Requirement (Maximum at Vehicle/TSM Interface)	552 (80.0)	538 (78.0)	538 (78.0)	632 (91.6)	616 (89.3)	616 (89.3)
Vehicle Heat Pressure (Height SSTO Liquid Level Below Shuttle ET Liquid Level = 36.3 m (119 ft))	0	0	0	-465 (-67.4)	-465 (-67.4)	-465 (-67.4)
Transfer Pump Discharge Pressure KPa (psig)	1432 (207.7)	961 (139.4)	1175 (170.4)	1271 (184.4)	645 (93.5)	883 (128.0)

determine if atm TP LOX can be pressurized with GO_2 the Tank Pressurization/Stratification program (Appendix A) was utilized. The program was input with data defining the existing LC 39 LOX storage tank and was run for the SSTO load sequence shown in Table III-2. The results are plotted in Figure III-3 and show that an ullage pressure of 170 KPa (10 psig) can be maintained with GO_2 pressurant. The GO_2 flowrate required to maintain this pressure, however, must be increased from 0.45 (1.0) to approximately 1.6 kg/sec (3.5 lb/sec) at 111°K (200°R).

In Figure III-3 it is also noted that the pressurization causes considerable heating of bulk liquid resulting in a 2.46×10^6 kg (5.36×10^6 lb) stratified layer of liquid at 66.1°K (119°R) on top of the bulk atm TP LOX. Since the tank is full of atm TP LOX at the beginning of loading, the enthalpy of the stratified layer has increased 19.6 kJ/kg (8.43 Btu/lb) or 47.7×10^6 kJ (45.2×10^6 Btu) were transferred into the tank liquid.

Figure III-3 also shows that approximately 1.4×10^6 kg (3×10^6 lb) of atm TP LOX remains in the tank at the termination of loading indicating that the tank is of sufficient capacity to support this loading sequence. Until depleted, the high density atm TP LOX will be delivered to the transfer line since it stratifies at the bottom of the tank.

2. Pump and Transfer Line

The temperature rise and corresponding density decrease of atm TP LOX between the storage tank and vehicle tank is caused by heat transferred to the fluid due to pump inefficiency and leakage through the pump casing; transfer line friction; and environmental heat leak through the line and other components.

The heat due to pump inefficiency was calculated with the $3.8 \text{ m}^3/\text{min}$ (1000 gpm) pump specification efficiency of 76% (Reference 25) and the capacities and pressure rise across the pump from Table III-3. The heat transfer through the casing was calculated by estimating pump surface area and heat transfer coefficients. The thermal energy added to the fluid by the pump during the transfer operations is presented in Table III-4.

The transfer line environmental heating was analyzed and presented in Appendix B. For the existing 15/20 cm (6/8-inch) diameter 565 m (1850-ft) long LOX transfer line, an environmental heat leak rate of 4.8 to 9.6 w/m of line (5 to 10 Btu/hr/ft of line) can be expected depending on the transfer duration.

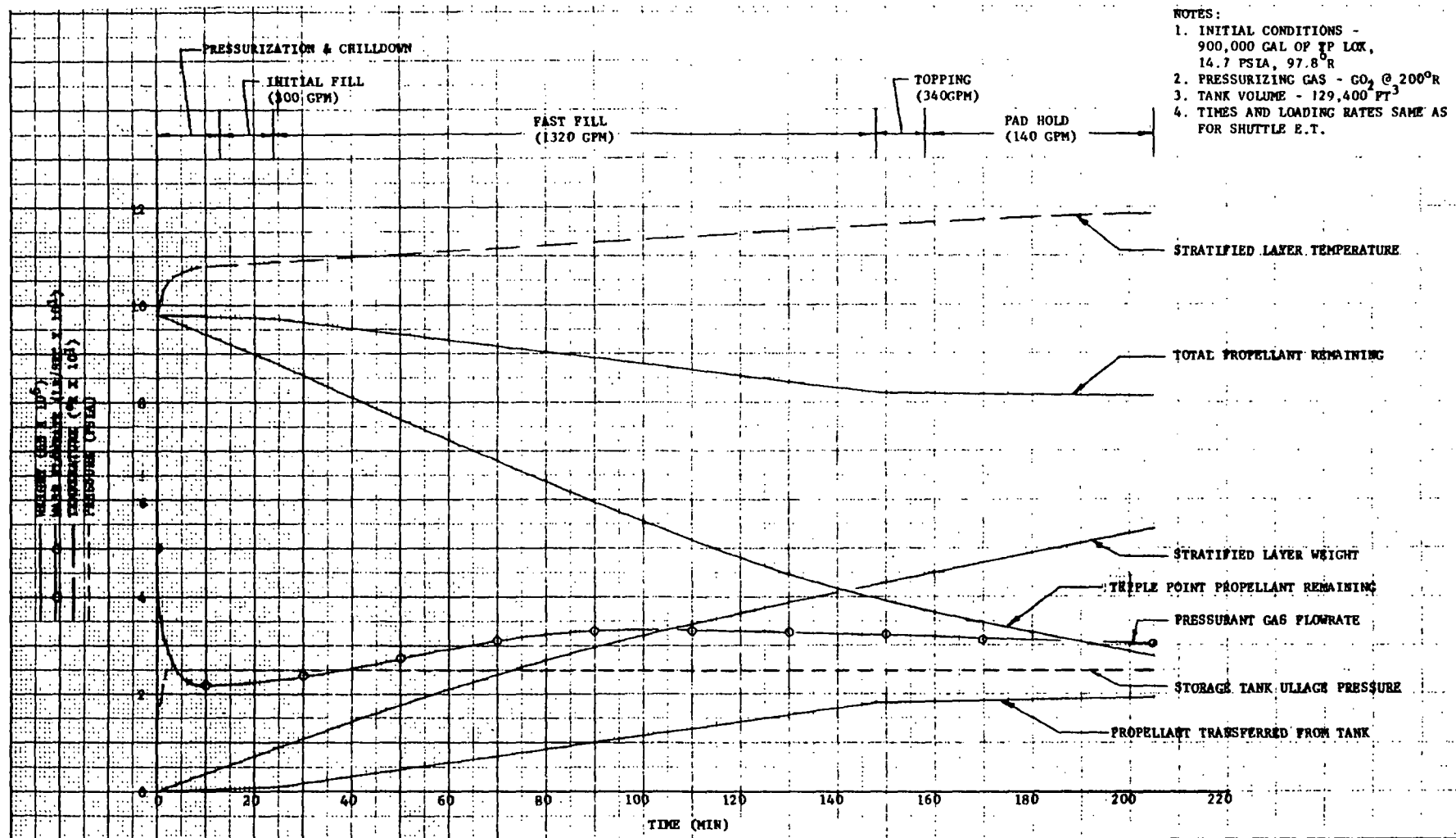


FIGURE III-3 SSTS/TP LOX PROPELLANT LOADING TIMELINE - STORAGE TANK PROFILE

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Table III-4. LOX Pump Heat Input to Fluid

Operation	Capacity m ³ /min (gpm)	Pressure Rise KPa (psi)	Q Pump, kj/kg (Btu/lb)	
			Pump Inefficiency	Casing
Initial Fill	3.14 (830)	515 (74.7)	0.123 (0.053)	0.026 (0.011)
Fast Fill	5.00 (1320)	1271 (184.4)	0.307 (0.132)	0.016 (0.007)
Topping	3.14 (830)	645 (93.5)	0.156 (0.067)	0.026 (0.011)
Replenish	3.14 (830)	882.5 (128.0)	0.214 (0.092)	0.026 (0.011)

The transfer line friction heating was calculated using the line pressure drops from Table III-3. The resulting heat inputs from the transfer line and pump are presented in Table III-5 and are shown parametrically for varying transfer line environmental heat leak rates.

The resulting temperature and density of sub-cooled LOX delivered to the vehicle tank are shown in Table III-6.

The tabulated results show that the average density of the fluid delivered to the vehicle is not sensitive to pump and line heating effects for the 9.6 w/in (10 Btu/hr-ft) insulated line and sub-cooled LOX at a density of 1306 kg/m³ (81.5 lb/ft³) can be delivered to the vehicle.

C. VEHICLE TANK LOADING

The SSTO vehicle configuration (Reference 1) provides two separate cylindrical tanks for the oxidizer. These tanks are made of 2219 aluminum alloy and form the load paths between the fuel tank and the engine mounts. The tank dimensions and other geometric information are shown in Figure III-4.

A thermal analysis of the SSTO TP LOX tank insulation was conducted and the results presented in Appendix D. The analysis concluded that from thermal as well as practical considerations an external foam insulation of 13 cm (1/2-inch) thickness should be used. This equates to a steady-state heat leak rate of 239 w/m² (76 Btu/ft²-hr) or 86.5 kw (2.95 x 10⁵ Btu/hr) into the two tanks.

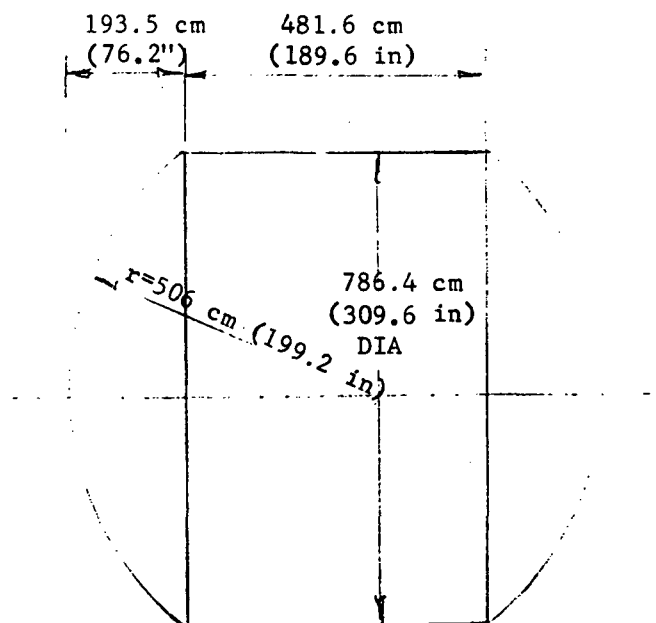
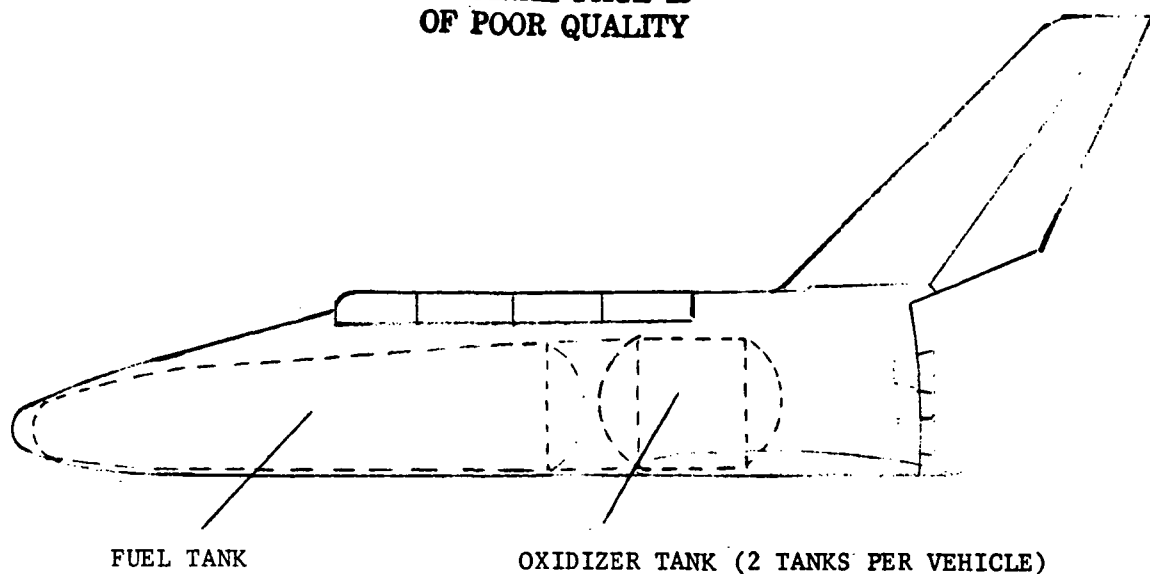
Table III-5. Pump and Transfer Line Heat Inputs

Operation	Load Rate m ³ /min (gpm)	Pump Q _p kJ/kg (Btu/lb)	Line Friction		Line Environmental		Q _{TOT} kJ/kg (Btu/lb)
			P _f KPa (psi)	Q _f kJ/kg (Btu/lb)	q _L w/m (Btu/hr-ft)	Q _E kJ/kg (Btu/lb)	
Initial Fill	1.14 (300)	0.149 (0.064)	43 (6.2)	0.033 (0.014)	9.6	0.221	0.402
					(10)	(0.095)	(0.173)
					38	0.888	1.070
					(40)	(0.382)	(0.460)
					77	1.777	1.958
Fast Fill	5.00 (1320)	0.323 (0.139)	666 (96.6)	0.509 (0.219)	(80)	(0.764)	(0.842)
					9.6	0.051	0.884
					(10)	(0.022)	(0.380)
					38	0.202	1.035
					(40)	(0.087)	(0.445)
Topping	1.29 (340)	0.181 (0.078)	55 (8.0)	0.042 (0.018)	77	0.405	1.237
					(80)	(0.174)	(0.532)
					9.6	0.195	0.419
					(10)	(0.084)	(0.180)
					38	0.784	1.007
Replenish	0.53 (140)	0.214 (0.092)	293 (42.5)	0.223 (0.096)	(40)	(0.337)	(0.433)
					77	1.568	1.791
					(80)	(0.674)	(0.770)
					9.6	0.477	0.700
					(10)	(0.205)	(0.301)
					38	1.905	2.342
					(40)	(0.819)	(1.007)
					77	3.810	4.247
					(80)	(1.638)	(1.826)

Table III-6. Temperature and Density of Propellant Delivered to SSTO/TP LOX Tank

Operation	Line Heat Leak, w/m (Btu/hr-ft)	Temp. Rise °K (°R)	Delivered Temp. °K (°R)	Density Decrease kg/m ³ (lb/ft ³)	Delivered Density kg/m ³ (lb/ft ³)	Density Change (% of TP Density)
Initial Fill	9.6 (10)	0.383 (0.690)	54.76 (98.56)	1.60 (0.10)	1305 (81.47)	0.12
+ Fast Fill	38 (40)	0.486 (0.874)	54.99 (98.74)	2.08 (0.13)	1305 (81.44)	0.16
+ Topping	77 (80)	0.622 (1.12)	54.99 (98.99)	2.56 (0.16)	1304 (81.41)	0.20
Replenish	9.6 (10)	0.284 (0.512)	54.66 (98.38)	1.12 (0.07)	1306 (81.50)	0.08
	38 (40)	1.27 (2.28)	55.64 (100.15)	7.85 (0.49)	1299 (81.08)	0.60
	77 (80)	2.41 (4.34)	55.64 (102.2)	13.0 (0.81)	1294 (80.76)	0.99

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VOLUMES PER TANK $-m^3 (FT^3)$

LO_2	322.8 (11401)
ULLAGE	10.0 (358)
INTERNAL STRUCTURE	5.1 (181)
TANK	338 (11940)

SURFACE AREA PER TANK $m^2 (FT^2)$

DOME	61.5 (662)
CYLINDRICAL SEC.	119 (1281)
TOTAL	180.5 (1943)

FIGURE III-4

SSTO OXIDIZER TANK CONFIGURATION

Since the insulation is external, the chilldown energy is essentially that required to cool the aluminum tank walls and structure. From Reference 1 the oxidizer tank weight is defined as 13.0 kg/m^2 (2.67 lb/ft^2) or 4706.5 kg ($10,376 \text{ lb}$) for both tanks. To cool this mass from ambient to triple point temperature $7.55 \times 10^5 \text{ kJ}$ ($7.16 \times 10^5 \text{ Btu}$) must be transferred into the propellant.

In order to maintain constant the average density in the vehicle tank after it is fully loaded, warm propellant from the top of the tank must be withdrawn and replaced with colder propellant in the bottom. This recirculation or pad hold flowrate is a function of the vehicle tank insulation and the ΔT of the inlet and outlet propellant and is shown parametrically in Figure III-5.

D. OTHER TP LOX CONSIDERATIONS

1. Thermal Contraction Effect on LOX System

The difference in temperature between NBP [-183.3°C (-297.9°F)] and TP [-218.8°C (-361.8°F)] LOX results in a significant difference in the contraction of materials used in the system. The resulting effects on the transfer line, storage tank, valves, pumps, expansion joints, flex hoses and other system components must be considered.

The longest section of LOX transfer line in the LC 39 system which is anchored at each end is 302.2 m (991.5 ft) consisting of 291.0 m (954.8 ft) of Schedule 5 Invar pipe, 2.51 m (8.25 ft) of Schedule 5 Type 304 stainless steel pipe and 8.66 m (28.4 ft) of Schedule 10 Type 304 stainless steel pipe. Upon cooling to -218.8°C (-361.8°F) and using thermal contraction data defined in Reference 30, a contraction of 16 cm (6.3 inches) was calculated. This results in a maximum tensile stress (in the Schedule 5 pipe) of 77.221 MPa ($11,200 \text{ psi}$) which is well within the minimum yield stress of the materials involved [344.7 MPa ($50,000 \text{ psi}$) for 304 stainless steel].

The LOX storage tank is 19.13 m (62 ft , 9-in) diameter, type 304 stainless steel sphere suspended inside a 20.96 m (68-ft , 9-in) diameter carbon steel outer shell. The inner sphere is supported concentrically within the outer shell by a system of vertical and horizontal rods. Since the inner sphere is essentially hung inside the outer shell, its contraction should not be of consequence. However, a detailed analysis of the inner tank supports, discharge and sensing lines would be required.

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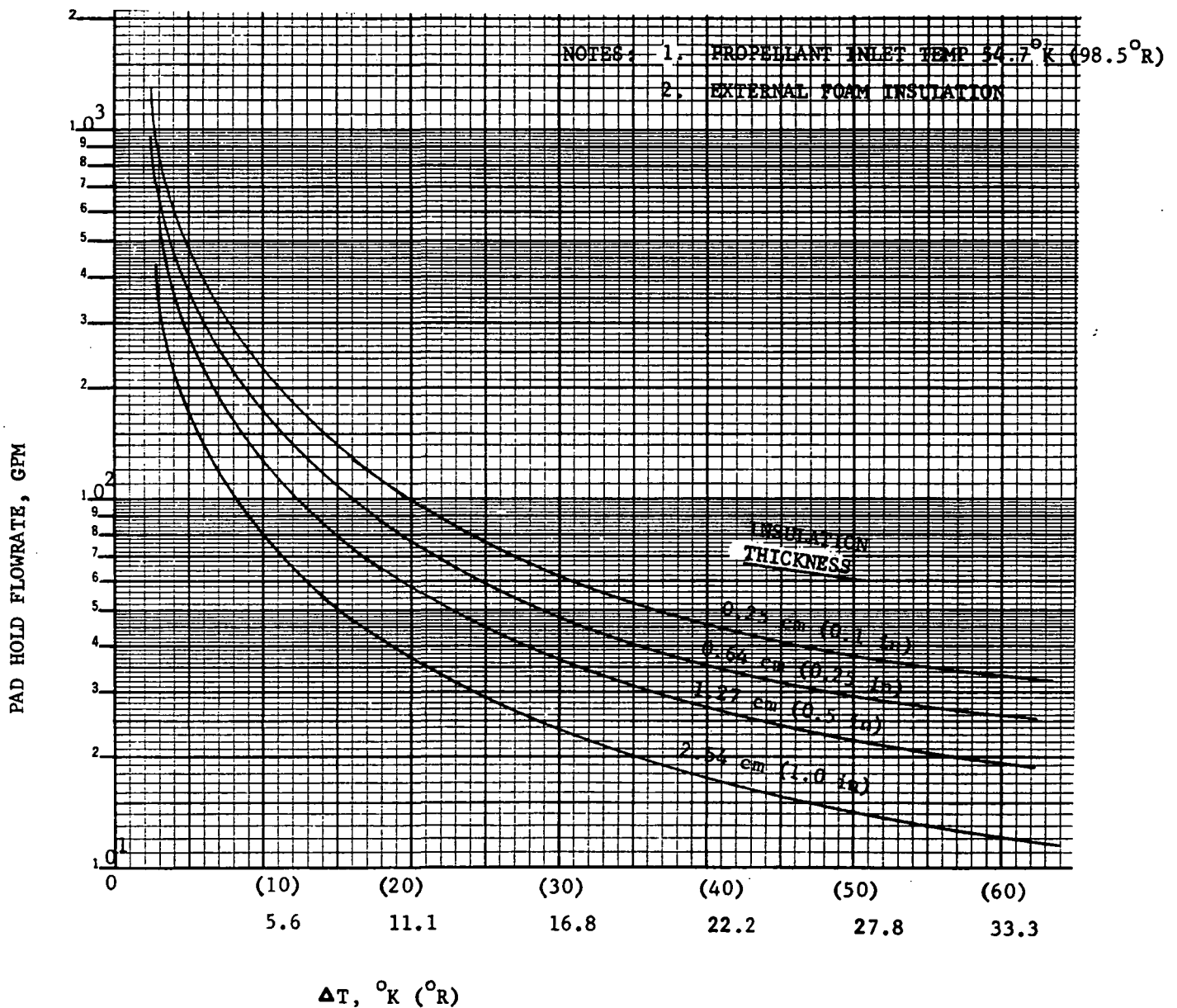


FIGURE III-5 SSTO/TP LOX PAD HOLD FLOWRATE VS PROPELLANT ΔT FOR VARYING INSULATION THICKNESS

A telecon survey was conducted in which the suppliers of the LOX storage and transfer components were requested to determine the effects of triple point LOX on their respective hardware. The firms contacted are listed below:

Byron/Jackson	Pumps
Pacific Valve	Gage Valves and Check Valves
Royal Industries	Pneumatic Butterfly Valves
Capital Westward	LOX Strainer and LOX Filter
Zallea Brothers	Expansion Joints
Anaconda Hose	Flex Hose Assembly
Masoneilan (Annin)	Pneumatic Flow Control Valves
Chemetron - Tube Turns Div.	Expansion Joint

All suppliers indicated that the hardware should be able to perform at triple point LOX temperatures and density. Only Zallea Brothers (expansion joints) expressed some concern due to increased joint shrinkage and resulting increased stresses at the lower temperature. They felt that the design is capable of withstanding the temperature but that cycle life, design margin, and other constraining parameters should be examined before a final commitment is made. These analyses should be conducted prior to utilization of TP LOX in the present system. However, no hardware changes are anticipated.

2. Cost of Triple Point Liquid Oxygen

Liquid oxygen for the shuttle program is produced by Air Products and Chemicals, Inc. at the Mims, Florida plant and is delivered to LC 39 at KSC in 18144 kg (20 ton) capacity roadable tankers. Through conversations with Air Products sales personnel it was learned that the present (1977) contract price to NASA is .077 \$/kg (.035 \$/lb) with no transportation charge since delivery distance is less than 50 miles.

To determine the cost of triple point liquid oxygen, the analytical technique used by Voth of the National Bureau of Standards to determine the cost of densified hydrogen was employed (Reference 20). This analysis includes an estimate of plant costs, input power costs, and operation and maintenance costs per unit of product oxygen. It does not include the cost of gaseous oxygen feed stock and costs are based on 1973 prices. To include the cost of oxygen feed stock and adjust to 1977 dollars the ratio of the analytical cost of TP LOX to NBP LOX was multiplied by the current cost of LOX to NASA. The resulting cost of TP LOX delivered to the launch site at KSC is .095 \$/kg (.043 \$/lb) (1977 dollars).

E. DENSIFIED OXYGEN SYSTEM CONCLUSIONS

The preceding analyses have evaluated the feasibility of storing and transferring triple point liquid oxygen in the existing KSC LC 39 LOX ground system. Since a pressure below atmospheric will collapse the storage tank, pressurization during standby periods must be provided. Due to the low temperature of TP LOX the pressurization and purge gas must be helium instead of nitrogen as presently used.

It was shown that a significant amount of heat is transferred into the propellant in the storage tank during loading but the tank capacity is sufficient to contain this heat in a stratified layer and still deliver triple point propellant to the transfer line. Also, the LOX vaporizer capacity must be increased from .45 to 1.6 kg/sec (1 to 3.5 lb/sec) in order to maintain the ullage pressure at 170 KPa (10 psig) for adequate pump NPSH. The heat input from the pump, transfer line and vehicle were shown to be minimal when compared to the heating from storage tank pressurization. A major unknown is the effect of vehicle tank pressurization and its contribution to the overall system heating.

This analysis, therefore, has shown that with the aforementioned modifications it is feasible to store and transfer TP LOX at the Shuttle loading rates with the existing LOX system.

For loading the SSTO tanks, however, further analysis is needed and a system defined for circulating liquid for upgrading and pad hold operations.

IV. DENSIFIED METHANE DISCUSSION

A. SLUSH METHANE STORAGE AND TRANSFER

Liquid methane has not been used to date in launch vehicle propulsion systems and, consequently, no vehicle loading requirements or ground support systems exist. Therefore, to establish a baseline system and loading requirements to evaluate the storage and transfer of slush methane (SLCH_4), the following assumptions were made:

- o The SSTO 100% SLCH_4 load is equal in weight to the Dual Mode, Series Burn SSTO 100% RP-1 Load [183,921 kg (405,476 lb)] defined in Reference 3, or 381.4 m^3 (100,755 gal) of SLCH_4 .
- o The SSTO SLCH_4 tank is the same as the Dual Mode Series Burn SSTO RP-1 tank except its height is 15.24m (50-ft) instead of 12.19m (40-ft) to account for the increase in volume due to density differential.
- o The ground storage and transfer system is the existing LC 39 LH_2 system.

These assumptions do not define an optimum SLCH_4 system but provide a basis for assessing the complexity of storing and transferring large quantities of SLCH_4 .

In addition to the lack of requirements, there is a significant lack of published data on the characteristics and physical properties of slush methane. The study of densified methane, therefore, should be considered as a preliminary order of magnitude analysis.

1. SLCH_4 Baseline System

The SSTO SLCH_4 baseline loading system is shown in Figure IV-1 and the baseline loading sequence in Table IV-1. This timeline was derived by using the established LH_2 system flow durations for the chilldown and replenish modes and the percent of load for each fill mode. The fast fill flowrate was calculated as the maximum achievable with the storage tank at maximum allowable pressure [722 KPa (90 psig)]. Slow fill rates were assumed to be 10% of the fast fill rate, and the replenish flowrate was assumed to be $0.38 \text{ m}^3/\text{min}$ (100 gpm). The quantity of 50% solid fraction SLCH_4

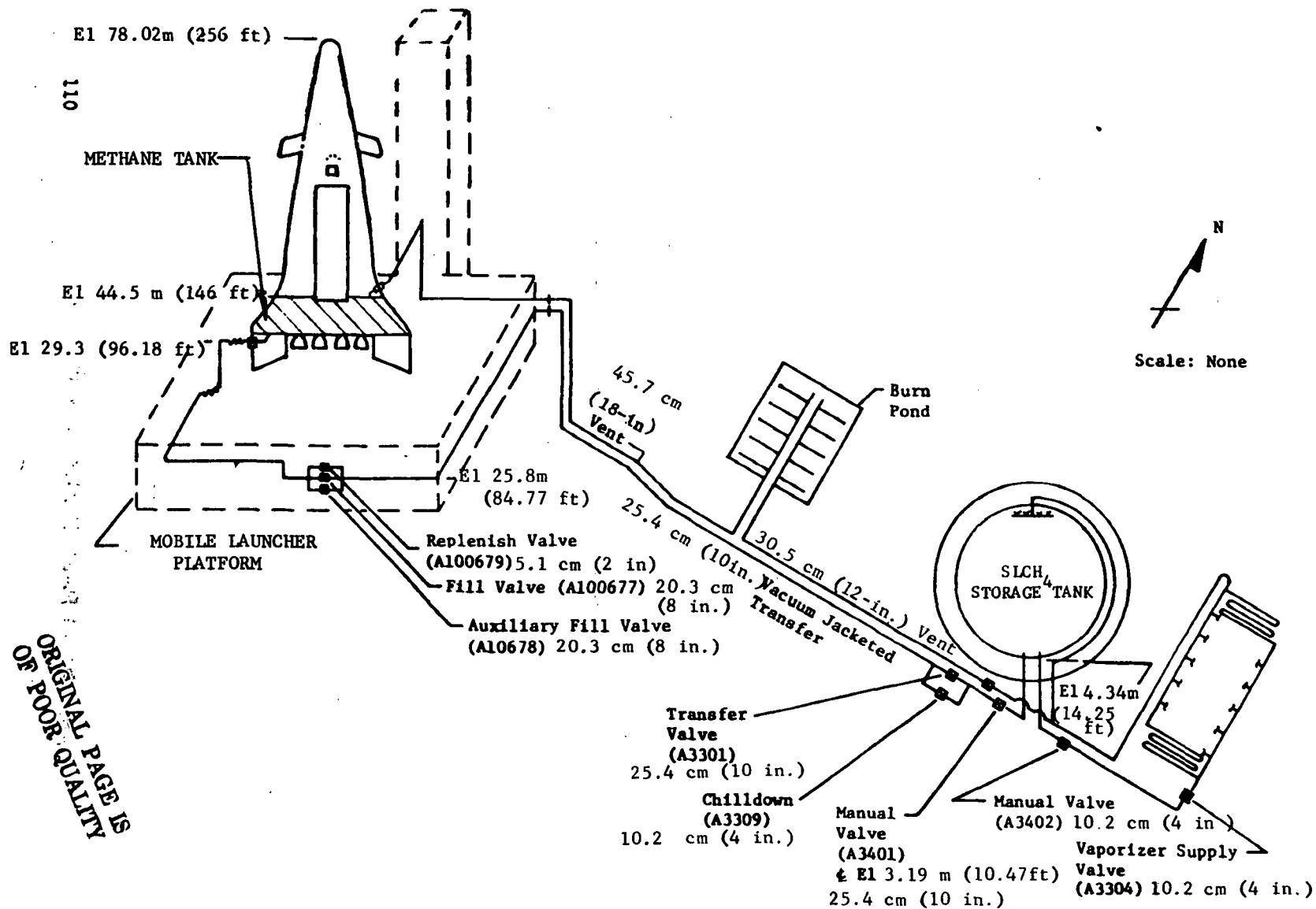


FIGURE IV-1 SSTO/DENSIFIED METHANE BASELINE LOADING SYSTEM

Table IV-1. SSTO SLCH₄ Baseline Loading Sequence

Operation	Time (Min.)	% Load	Load Rate m ³ /min (gpm)	Quantity m ³ (gal)
Facility & Vehicle Chillo down	10.0	0	0.43 (113)	4.28 (1,130)
Storage Tank Pressurization	TBD		TBD	45.05 (11,900)
Initial Fill	4.0	0-2	1.89 (500)	7.63 (2,015)
Fast Fill	19.3	2-98	18.9 (5,000)	366.14 (96,725)
Topping	4.0	98-100	1.89 (500)	7.63 (2,015)
Replenish	45.0	100	0.38 (100)	17.03 (4,500)
Total	82.3			447.76 (118,285)

for facility and vehicle chillo down was calculated by using the LH₂ system chillo down requirement converted to 50% SLCH₄ heat capacity, density, and temperature. The quantity required for storage tank pressurization and expulsion was determined as the displaced volume with the ullage gas initially at triple point conditions and finally at -73°K (-100°F) and 722 KPa (90 psig). Table IV-1 shows that vehicle fill to 100% is accomplished in 37.3 minutes and the quantity of SLCH₄ required for SSTO loading is 447.76 m³ (118,285 gal). This analysis does not consider tank stratification or solid melting as in the SLH₂ analysis.

2. Storage

The LH₂ storage tanks were designed and constructed to the requirements of Section VIII of the ASME Pressure Vessel Code for a 722 KPa (90 psig) internal working pressure (Reference 6). The tank is insulated by a 0.9m (3-ft) annular space between inner and outer tanks filled with perlite and evacuated to a pressure <500 microns. The tank liquid capacity is 3217.6 m³ (850,000 gal) of NBP LH₂ with a 10% ullage. Considering the maximum allowable capacity of the tank to be 3217.6 m³ (850,000 gal) at NBP LH₂ density [70.8 Kg/m³ (4.42 lb/ft³)], then the maximum allowable capacity of SLCH₄ at a density of 482 kg/m³ (30.1 lb/ft³) would be 472.4 m³ (124,800 gal) with a minimum ullage volume of 3102.5 m³ (819,600 gal). This computes to an 87% ullage volume and 5% margin in capacity over that required for an SSTO loading thus rendering the use of the

LH₂ tank for SLCH₄ not practical. It is therefore concluded that a tank of similar construction but sized for the specific mission requirements would be required.

Storage of SLCH₄ at triple point pressure [11.7 KPa (1.7 psia)] in the LH₂ or a similarly constructed tank can be allowed as the tank was designed to withstand a full vacuum in the inner sphere. The hazard of air leakage into the low vapor pressure LCH₄ is minimal since the annular space is maintained at a pressure less than the inner sphere. An inert gas blanket pressure must be maintained in the storage tank fill, discharge and vent lines and an inert environment maintained around sampling and sensing lines and other tank protrusions similar to that discussed for the LH₂ system. During transfer operations no air can enter the system as the entire system pressure is above 1 atm.

Quality decay of stored methane in a tank built and insulated similar to the LH₂ tank will be acceptable since it has been shown that slush hydrogen with the same heat of fusion can be adequately stored at a much lower temperature [13.8°K (24.9°R)].

3. Transfer

The LC 39 LH₂ transfer system consists of approximately 520 m (1700 ft) of 24 cm (10-inch) diameter line, 15.24m (50 ft) of 20 cm (8-inch) diameter line, three valves, two flex hoses and numerous joints and protrusions between the storage tank and Vehicle/Tail Service Mast (TSM) interface. A pressurized transfer of SLCH₄ with the storage tank at its maximum operating pressure [621 KPa (90 psig)] and the vehicle tank orientation as shown in Figure IV-1 yields a maximum flowrate of 18.9 m³/min (5000 gpm). By utilizing this value as the fast fill rate and the other rates defined by the Loading Sequence (Table IV-1), the transfer line pressure drop for various vehicle fill rates was calculated and is shown in Table IV-2. For the ground system from storage tank outlet valve A3301 to the vehicle/TSM interface, the pressure drop attributable to line friction loss was calculated using parameters defined previously for the slush hydrogen system (Paragraph II.A.) and SLCH₄ fluid properties (Reference 31). The vehicle pressure drop was assumed to be 34 KPa (5 psia) at 18.9 M³/min (5000 gpm) since no comparable system exists. No pressure drops are included for filters since slush fluids cannot be filtered with effective sized devices.

It should be noted that research to date has not defined an acceptable inert gas to use for blanket pressure, purging, or pressuriza-

Table IV-2. LC 39 LH₂ Transfer System Pressure Drop vs Flowrate for 50% Slush Methane

Component	Pressure Drop KPa(psid)			
	Slow Fill 1.9 m ³ /min (500 gpm)	Fast Fill 1.9 m ³ /min (500 gpm)	Topping 1.9 m ³ /min (500 gpm)	Replenish 0.38 m ³ /min (100 gpm)
Transfer Line Pressure Drop (Piping and Components)	497.7 (72.18)	390 (56.55)	425.6 (61.73)	425.9 (61.77)
GSE Head Pressure (Storage Tank to SSTO/TSM Interface) 26.2m (86 ft)	122.5 (17.77)	122.5 (17.77)	122.5 (17.77)	122.5 (17.77)
Total GSE Pressure Drop	620.1 (89.95)	512.4 (74.32)	548.1 (79.50)	548.5 (79.55)
SSTO Fill Line Pressure Drop (Piping and Components)	0.34 (0.05)	36.1 (5.23)	0.34 (0.05)	
SSTO Tank Head Pressure 15.24 m (50 ft)	0 (0)	72.05 (10.45)	72.05 (10.45)	72.05 (10.45)
Total Pressure Drop	621 (90.0)	621 (90.0)	621 (90.0)	621 (90.0)

Table IV-3. SLCH_4 Slush Quality Change in Transfer Line

Operation	Load Rate (gpm)	Time (min)	SLCH_4 Mass kg (lb)	P_{friction} KPa (psi)	Q_{friction} Kj (Btu)	Line Heat Leak		Q_{TOT} Kj (Btu)	Quality Change X (%)
						(Btu/hr-ft)	Kj (Btu)		
Slow Fill	1.89 (500)	4	3642 (8049)	498 (72.2)	3770 (3573)	9.6 (10)	1231 (1167)	5001 (4740)	2.28
						38 (40)	4924 (4667)	8693 (8240)	3.97
						77 (80)	9846 (9333)	13,616 (12,906)	6.21
Fast Fill	18.9 (5000)	19.3	175,765 (388,352)	390 (56.5)	142,345 (134,924)	9.6 (10)	5939 (5629)	148,283 (140,553)	1.40
						38 (40)	23,754 (22,516)	166,100 (157,441)	1.57
						77 (80)	47,509 (45,032)	237,366 (224,991)	2.24
Topping	1.89 (500)	4	3643 (8049)	425 (61.7)	3222 (3054)	9.6 (10)	1231 (1167)	4453 (4221)	2.03
						38 (40)	4924 (4667)	8146 (7721)	3.72
						77 (80)	9846 (9333)	13,068 (12,387)	5.96
Total			183,050		149,336	9.6 (10)	8401 (7963)	157,737 (149,514)	1.43
						38 (40)	33,603 (31,851)	182,939 (173,402)	1.66
						77 (80)	67,208 (63,704)	264,050 (250,284)	2.39

tion of liquid methane (Reference 32). This was further corroborated through conversations with cryogenics personnel at the National Bureau of Standards.

Absorption of most common inert gases, including nitrogen and helium into liquid methane is high. Storage tank pressurization can be achieved by vaporizing SLCH_4 , as has been demonstrated for subcooled hydrogen, but pressurization of the vehicle tank to prevent its implosion is a problem that must be resolved.

The degradation of slush quality between the storage and vehicle tanks in the baseline system was investigated. The friction heat inputs to the transferred fluid were derived with the line pressure drop values previously discussed for slush hydrogen. Values of 35 to 277 Kj/hr per lineal meter (10 to 80 Btu/hr per lineal foot) of line were used as heat rates from the environment to the fluid.

The results are presented in Table IV-3, where the sources and the amount of heat input to the fluid, and the corresponding slush degradation are listed for the slow fill, fast fill and topping operations. It has been shown that the average environmental heat leak for the existing LH_2 transfer line is approximately 35 Kj/hr-m (10 Btu/hr-ft) (Appendix B). This corresponds to a total quality decay in the SLCH_4 system transfer line of 1.43%.

For the replenish mode of $.38 \text{ m}^3/\text{min}$ (100 gpm), the quality degradation varies from 4.2 to 24% for line environmental heat leakage rates between 9.6 to 77 W per lineal meter (10 and 80 Btu/hr per lineal foot) of line. However, since the replenishing flowrate required to maintain the desired slush quality in the vehicle tank does not depend only on degradation during transfer, the complete evaluation of this mode must be made in conjunction with a vehicle tank insulation study.

B. TRIPLE POINT LIQUID METHANE (TP LCH_4) STORAGE AND TRANSFER

To assess the ground support requirements for a triple point liquid methane-fueled SSTO the same approach and assumptions were used for the slush methane system (Paragraph IV.A.)

1. TP LCH_4 Baseline System

The SSTO TP LCH_4 baseline loading system is the same as the slush methane system shown in Figure IV-1 and the baseline loading sequence in Table IV-4. This timeline was devised by using the

Table IV-4. SSTO TP LCH₄ Baseline Loading Sequence

Operation	Time (Min)	% Load	Load Rate m ³ /min (gpm)	Quantity m ³ (gal)
Facility & Vehicle Chillydown	10.0	0	0.519 (137)	5.19 (1,370)
Storage Tank Pressurization	TBD	0-100	TBD	48.1 (12,700)
Initial Fill	4.3	0-2	1.90 (500)	8.14 (2,150)
Fast Fill	20.6	2-98	19.0 (5000)	390.8 (103,239)
Topping	4.3	98-100	1.90 (500)	8.14 (2,150)
Replenish	45.0	100	0.38 (100)	17.0 (4,500)
Total	84.2			447.4 (126,109)

load rates established for the SLCH₄ baseline loading sequence. The quantity of TP LCH₄ for facility and vehicle chillydown was calculated by using the LH₂ system chillydown requirement converted to TP LCH₄ heat capacity, density and temperature. The quantity required for storage tank pressurization and expulsion was determined as the displaced volume with the ullage gas initially at triple point conditions and finally at -73°K (-100°F) and 722 KPa (90 psig). Table IV-4 shows that vehicle fill to 100% is accomplished in 39.2 minutes and the quantity of TP LCH₄ required for an SSTO loading is 477.37 m³ (126,109 gal).

2. Storage

The rationale for storage of slush methane in the LC 39 LH₂ tanks similarly applies to the storage of triple point liquid methane. The maximum allowable volume of TP LCH₄ would be 504.2 m³ (133,200 gal) resulting in an 86% ullage volume and 5% margin in capacity which are impractical limits. A tank of similar structural and thermal construction but sized for the specific mission requirements for TP LCH₄ would be required.

Density decay of stored TP LCH₄ in a tank built and insulated similar to the LH₂ tank will be acceptable since it has been shown that TP LH₂ can be adequately stored in this tank at a much lower temperature [13.8°K (24.9°R)].

3. Transfer

By utilizing the TP LCH₄ baseline loading system and sequence previously discussed the system pressure drop at varying vehicle fill rates was calculated (Table IV-5). For the ground system from storage tank valve A3301 to the vehicle/TSM interface, the pressure drop resulting from line friction loss was calculated using parameters previously defined for the triple point hydrogen system (Paragraph II.B.) and TP LCH₄ properties (Reference 32).

Table IV-5. LC 39 LH₂ Transfer System Pressure Drop vs Flowrate for TP Liquid Methane

	Pressure Drop, KPa (psi)			
	Slow Fill 1.89 m ³ /min (500 gpm)	Fast Fill 18.9 m ³ /min (5000 gpm)	Topping 1.89 m ³ /min (500 gpm)	Replenish 0.38 m ³ /min (100 gpm)
Transfer Line Pressure Drop (Piping and Components.	502.4 (72.86)	401.4 (58.22)	434.9 (63.07)	435.2 (63.12)
GSE Head Pressure (Storage Tank to SSTO/TSM Interface) 26.2 m (86 ft)	116.1 (16.84)	116.1 (16.84)	116.1 (16.84)	116.1 (16.84)
Total GSE Pressure Drop	618.5 (89.70)	517.5 (75.06)	551.0 (79.91)	551.3 (79.96)
SSTO Fill Line Pressure Drop (Piping and Components)	0.34 (0.05)	33.8 (4.90)	0.34 (0.05)	0.0
SSTO Tank Head Pressure [15.2 m (50 ft)]	0	67.5 (9.79)	67.5 (9.79)	67.5 (9.79)
Total Pressure Drop	618.8 (89.75)	618.8 (89.75)	918.8 (89.75)	618.8 (89.75)

The analysis of the triple point liquid methane transfer was based on the assumptions that fluid is at TP temperature and at storage tank pressure [618 KPa (90 psig)] at the start of transfer after system cooldown. The same range of line environmental heat leak

rates as for the SLCH₄ system was used to evaluate the temperature and density changes of the transferred fluid. The results are summarized in Table IV-6.

Table IV-6. Average Temperature and Density Change of TP LCH₄ in Transfer Line

Operation	\dot{q}_L w/min (Btu/hr-ft)	Average Temp. Change °K (°R)	Density kg/m ³ (lb/ft ³)	Density Change (% of TP Density)
Slow Fill +	9.6 (10)	0.622 (1.20)	450.8 (28.14)	0.18
Fast Fill +	38 (40)	0.667 (1.20)	450.6 (28.13)	0.19
Topping	77 (80)	0.806 (1.45)	450.4 (28.12)	0.23
Replenishing (100 gpm)	9.6 (10)	1.14 (2.05)	450.1 (28.10)	0.32
	38 (40)	2.75 (4.95)	447.9 (27.96)	0.81
	77 (80)	4.89 (8.81)	445.0 (27.78)	1.40

The results indicate that, within the heat leakage rates investigated, the transfer of TP methane does not present any problem. As in the case of the slush methane, the replenish flow impact on the overall vehicle load is a function of vehicle tank insulation.

C. DENSIFIED METHANE CONCLUSIONS

The preceding analyses have shown that a ground storage and transfer system of similar design to the LC 39 LH₂ system can support the loading of the SSTO vehicle with both slush and triple point liquid methane. The vacuum-jacketed storage tank affords adequate insulation and the enthalpy rise in the multilayer insulated vacuum jacketed transfer line is minimal. It also appears feasible to transfer densified methane by pressurized tank expulsion as in the case of LH₂.

Since this analysis was intended to provide a basis for assessing the complexity of storing and transferring large quantities of densified methane, no attempt was made to optimize the system design. Trade studies of storage tank insulation, pressure vs pump transfer, transfer line insulation, and vehicle tank insulation similar to those performed for the densified hydrogen system must be conducted to adequately define the densified methane system design requirements.

V. OTHER DENSIFIED PROPELLANT CONSIDERATIONS

A. GROUND SYSTEM INSTRUMENTATION

Instrumentation presently in use in the LC 39 LH₂ and LOX ground systems measures the parameters of pressure, temperature, fluid quality, liquid level, and flowrate. System static pressures are measured with transducers located a sufficient distance from the cryogenic fluid so as not to be effected by the temperature. Some system pressure gages and transducers will not measure below atmospheric pressure and must be changed to absolute devices where triple point pressures occur. The existing type of pressure instrumentation, therefore, should function equally well for slush or triple point propellant as for normal boiling point propellant with the aforementioned changes.

Instruments for measuring temperature in the LC 39 LH₂ and LOX systems are of the resistance type (Reference 33). For the LH₂ system a probe with a Germanium crystal resistor is located at the point where the temperature is to be measured. A carbon crystal resistor is used in the LOX system. The temperature is determined by measuring the current flow through the element whose resistance is a function of temperature. The temperature probes in the LC 39 LH₂ and LOX systems are supplied by Scientific Instruments, Inc., of Lake Worth, Florida. Conversations with Mr. Hoey of Scientific Instruments confirm that the range of temperature measurement with these devices is only a matter of calibration to the end points desired and that performance will not be effected at triple point LH₂ and LOX temperatures.

Fluid quality is measured by a similar type resistance element device called discrete liquid sensor and indicates whether liquid is or is not present (Reference 34). As liquid comes in contact with the element, a change in resistance is detected. These instruments are also supplied by Scientific Instruments and, according to Mr. Hoey, their use in either slush or triple point liquid will not effect performance, only calibration. This instrument will not, however, provide information as to the quality of slush being transferred.

The liquid level and flowrate instrumentation in the existing systems is simply a measure of pressure. The liquid level gages measure the liquid pressure head with the output calibrated in volume (gallons). The flowrate measurement devices are a differen-

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121

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tial pressure measurement across an in line venturi or orifice. As previously discussed, existing pressure transducers will not be effected if used for triple point of slush propellant service.

In the case of slush hydrogen and methane, a new measuring device will be required to determine the mass of propellant in the storage tank. Due to the characteristics of slush, the solid particles will settle to the bottom of the tank resulting in a non-homogeneous density. If the average density in the tank were known the volume and mass of propellant in the tank could be determined by dividing the density into the liquid head pressure. Recent conversations with R. S. Collier of the National Bureau of Standards, Boulder, Colorado, have revealed the development of a radio frequency density measuring device that has been successfully tested in slush hydrogen. Per Mr. Collier, the device measures the resonant frequency of the slush container which is a function of the average dielectric constant of the tank, and the output is calibrated in average fluid density. This device in conjunction with the existing liquid level system would define the total mass and average slush quality of propellant in the tank.

B. GROUND TRANSPORTATION OF DENSIFIED LH₂ AND LOX

To assess the feasibility of transporting densified LH₂ and LOX, personnel in the Air Products and Chemicals, Inc. Engineering Department were contacted.

LH₂ and LOX are presently delivered to the LC 39 storage tanks in roadable tankers. The LH₂ tankers deliver approximately 49 m³ (13,000 gallons) manufactured by Air Products and Chemicals, Inc. in New Orleans, Louisiana. If used for the transporting of slush and triple point hydrogen, these tankers could only be filled to 42.6 m³ (11,250 gal) with SLH₂ and 45.2 m³ (11,950 gal) with TP LH₂ due to their increased densities. Transfer of LH₂ from the tanker to the storage tank is accomplished by a pressurized discharge with ullage pressure supplied from vaporized LH₂.

The LOX tankers deliver approximately 20.5 m³ (5400 gal) manufactured by Union Carbide, Linde Division in Mims, Florida. The maximum allowable volume of TP LOX that could be transported due to increased density is 17.8 m³ (4700 gal). Transfer of LOX from the tanker is accomplished by pump discharge with pump head pressure supplied from vaporized LOX.

In each case, the tanker design allows for a minimum of 1 atm pressure in the inner tank which precludes the transporting of LH₂ and LOX at triple point pressure. The rate of LH₂ boiloff in the hydrogen tankers is 0.5% per day which equates to a heat leak of 90 w (7400 Btu/day). Considering this heat leak, the hydrogen latent heat of fusion and the tanker filled to its maximum SLH₂ capacity, a slush quality decay of 4% per day was calculated. For a full load of TP LH₂ and the same heat leak a density decay of 0.2307 kg/m³ (0.0144 lb/ft³) per day was calculated. Provisions also exist for delivery of LH₂ to LC 39 in 106 m³ (28,000 gal) rail car dewars with a boiloff rate of 0.3% per day. For a maximum allowable load of 91.8 m³ (24,250 gal) of SLH₂ a quality decay of 2.3% per day can be expected, and for a maximum allowable load of 97.4 m³ (25,730 gal) of TP LH₂ a density decay of 0.0897 kg/m³ (0.0056 lb/ft³) per day can be expected.

The boiloff rate for the LOX roadable tanker is 2% per day and equates to a heat leak at the triple point temperature of 1345 w (110,108 Btu/day). The resulting density decay of a full load of TP LOX is 37.27 kg/m³ (2.327 lb/ft³) per day due to the proximity of the Manufacturing Facility to the launch site. This is not considered a major problem.

It is feasible, therefore, to transport densified LH₂ and LOX in the existing roadable tankers if modifications to insure a positive tank pressure are made. For the subcooled hydrogen system, however, it has been shown that the liquefaction and densification plants should be integrated with the loading system thus precluding its need for transportation.

VI. CONCLUSIONS

This study has evaluated the ground and vehicle system requirements for loading an SSTO vehicle with densified propellants using the existing Shuttle systems at KSC as a baseline. Since the industry interest during this study centered more on densified hydrogen than oxygen or methane, and considerably more experimental data is available on densified hydrogen, the analysis of slush and triple point liquid hydrogen was much more extensive.

Specific conclusions relevant to each propellant are included at the end of their respective sections. For densified hydrogen (Paragraph II.F), the advantage of using slush over triple point liquid was clearly shown to the extent that future considerations should be directed toward slush alone.

For triple point liquid oxygen (Paragraph III.E), the feasibility of using the existing KSC system was shown with certain reservations and the addition of a recirculation system for upgrading and pad hold.

The analysis of densified methane (Paragraph IV.C), both slush and liquid, indicated that a system of similar design to the existing liquid hydrogen ground system could be used for loading the SSTO.

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125

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VII. RECOMMENDATIONS

The densified hydrogen analyses were based primarily on small scale experimental test data published by the National Bureau of Standards. Furthermore, the tank pressurization analysis, which is a key element of this study, was based on a NBP cryogenic liquid model applied to slush and triple point liquid. Therefore, to validate the model and these data for vehicle applications, large scale experimental testing of densified hydrogen (larger scale than NBS testing) must be performed to validate the results of this study. This testing is also needed to establish design criteria for slush mixing in large storage tanks and slush fluidizing velocities in large lines.

Further densified hydrogen development and analysis in the areas of storage tank pressurization versus pump transfer and vehicle tank insulation are recommended since these are the prime contributors to system enthalpy gain. Also, development is required in the areas of slush filtration and instrumentation for measurement of the average densities of densified hydrogen.

Since this study did not develop the densified liquid oxygen system to the extent that was done for densified liquid hydrogen, further analytical studies in this area are in order. The system of analysis developed for hydrogen can be applied to densified oxygen and the optimum loading system defined. This study evaluated only subcooled liquid oxygen but, in view of the significant ground system advantages shown for slush over densified liquid hydrogen, slush oxygen should be analyzed as well. Also, since experimental data of densified oxygen characteristics and producibility are almost nonexistent, further efforts in this area should include development testing.

The method of analysis for hydrogen applies equally to slush and triple point liquid methane systems and should be performed if interest in densified methane propellant continues. Also, experimental test data of densified methane characteristics and producibility are needed.

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127

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APPENDIX A

CRYOGENIC TANK PRESSURIZATION/STRATIFICATION ANALYSIS MODEL

The analyses of both the propellant ground storage tank pressurization and outflow process and the SSTO propellant tank loading process were conducted using the Martin Marietta tank pressurization program. This program is an outgrowth of the model originally developed by Morey and Traxler as described in Chapter 18 of Reference 35. Modifications have been made to this original model to accommodate cryogenic propellants and insulated tank walls. An improved treatment of liquid/gas interface heat and mass transfer together with an empirical Nusselt versus Reynolds number correlation for heat transfer resulting from pressurant inflow has also been incorporated into the model.

The program predicts time histories of tank pressure, ullage gas and liquid temperature, tank wall temperature, gas and liquid masses, and pressurant usage requirements. Pressurants may be either condensible or non-condensible. Various options in the program permit the analysis of regulated, blowdown, and venting processes. In addition, the program is flexible so that the analysis of both draining and filling of a propellant tank is possible. A stratification model developed by T. E. Bailey, R. VandeKoppel and G. Skartvedt was adapted for use with slush propellants and incorporated into the program. This model is described in Chapter 12 of Reference 35.

The tank model used in the analysis is shown in Figure A-I. A mass and energy balance is conducted on the vapor in the ullage in order to determine pressure, P_G , temperature, T_G , and mass, W_G , as functions of time. Energy and mass are introduced into the ullage during pressurization or lost from the ullage during venting. A warm liquid layer exists at the interface between the ullage and the bulk liquid (slush or triple point) region in the tank. This liquid layer results from condensation of the ullage gas and from stratification of the liquid in the bulk region. Gas condensation is calculated at both the wall, \dot{W}_{CW} , and at the liquid-gas interface, \dot{W}_{CU} . The average temperature of the upper layer, T_{UL} , is calculated from an energy balance that considers heat transfer from the ullage to the layer, \dot{Q}_{GU} , and from the upper layer to the bulk region, \dot{Q}_{UB} . Heat lost from the ullage to the tank wall, \dot{Q}_W , and that transferred into the bulk region, \dot{Q}_{WL} , are also calculated. The effects of tank wall heat capacity are included in the heat transfer calculations. Additional assumptions employed in the model include:

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129

TANK PRESSURIZATION/
STRATIFICATION MODEL

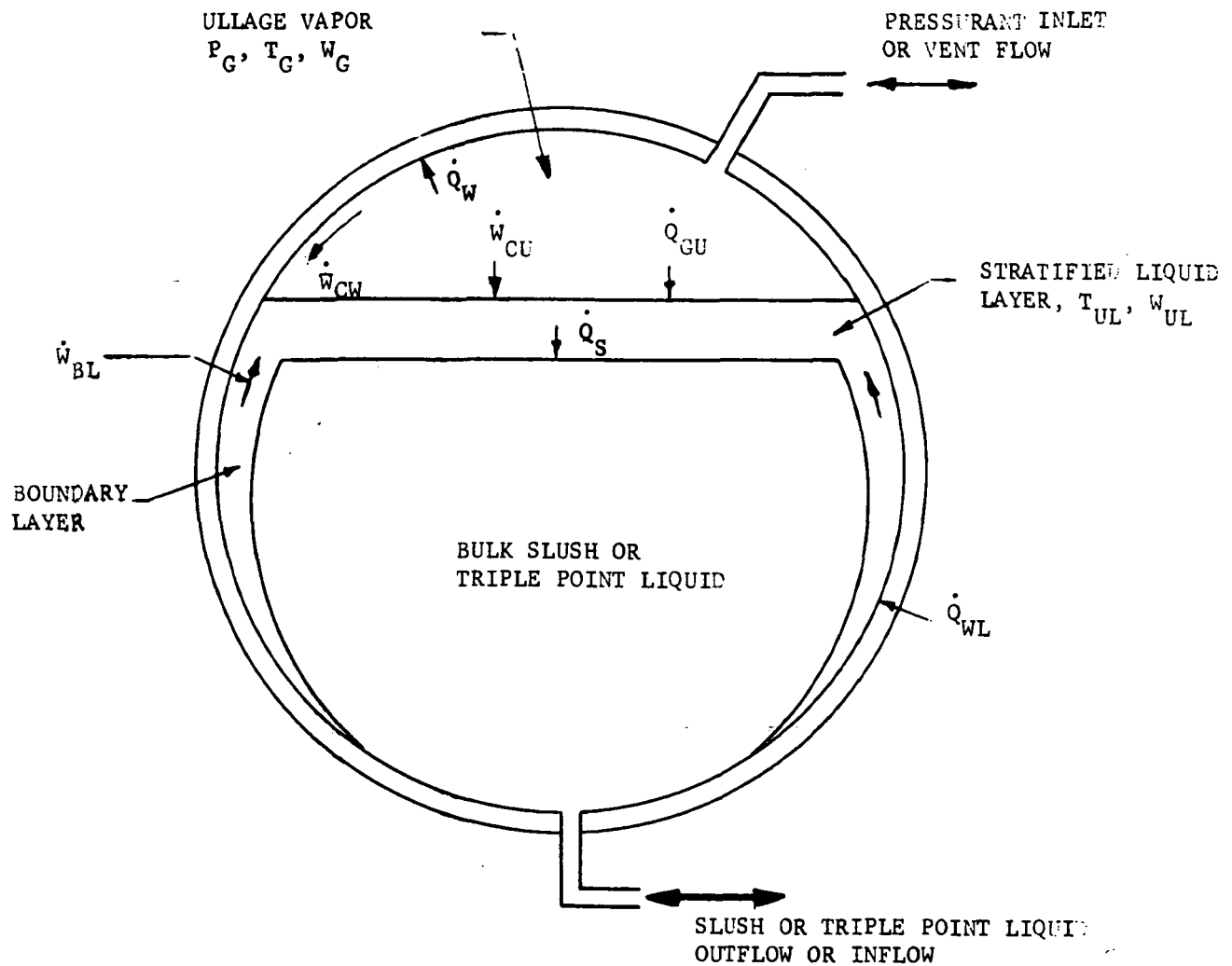


FIGURE A-1

- 1) Temperature and density distribution in the ullage and bulk liquid (slush) regions are uniform.
- 2) All of the wall heat input goes into a wall boundary layer, which remains attached to the wall.
- 3) The wall boundary layer, and the resulting heat transfer coefficient, are the same as would be found with a vertical plate in a pure liquid (this amounts to assuming that the solid near the wall is melted back to a distance at least equal to the boundary layer thickness, and also that the deviations from verticality have an insignificant effect on the boundary layer flow).
- 4) The boundary layer is very thin, so that the thermal energy content of this layer can be neglected.
- 5) The surface to volume ratio of the solid phase is very high, so thermal equilibrium always exists between the solid and its interstitial liquid.
- 6) The vapor/liquid interface is saturated.
- 7) The heat transfer coefficients from the vapor to the vapor/liquid interface and from the interface to the stratified layer are given by free-convection correlations for a horizontal surface with a stable gradient.
- 8) The boundary layer flow into the stratified layer and the condensation at the interface, mix together to give a uniform stratified layer temperature except near the vapor and slush interfaces.

Input requirements include pressurant and liquid thermophysical properties; control parameters such as regulator and vent pressure settings; and tank geometric data such as volume versus height, and wetted surface area versus liquid volume. Operational input requirements include initial tank pressure, gas and liquid temperatures, initial ullage, heat transfer into the tank from the environment, and liquid inflow or outflow rates. The control and operational input parameters may be varied as functions of time.

APPENDIX B

TRANSFER LINE THERMAL ANALYSIS

This section presents the analysis performed to evaluate the environmental heating of the transfer lines for liquid hydrogen and oxygen at triple point temperature. The analysis was performed for the basic straight piping in three different configurations: vacuum jacketed line, vacuum jacketed with multilayer insulation (MLI) in the annular space, and with active cooling.

Vacuum Jacketed Line Without MLI

The vacuum jacketed line without MLI consists of two concentric pipes, with the annulus between the pipes evacuated to provide an efficient barrier for the transfer of the heat. The basic line section is sketched in Figure B-1 with the thermal network that simulates the heat transfer process.

For the analysis, it is assumed that a vacuum of 10^{-5} torr or better exists in the annular space so that the heat transfer by gas conduction may be neglected. The heat transfer by radiation between the outer and the inner pipe is given by:

$$Q = A_i \bar{F}_{i-o} \sigma (T_o^4 - T_i^4) \text{ Btu/hr-ft} \quad (B1)$$

where A_i = surface area of one linear foot of inner line (ft^2)

\bar{F}_{i-o} = radiation interchange factor

σ = Stefan-Boltzman constant

T_o, T_i = outer, inner pipe temperature ($^{\circ}\text{R}$).

The radiation exchange factor \bar{F}_{i-o} for two concentric cylinders is:

$$\bar{F}_{i-o} = \frac{\epsilon_i \epsilon_o}{\epsilon_o + \frac{A_i}{A_o} (1 - \epsilon_o)} \quad (B2)$$

where ϵ is the pipe surface emissivity and the subscripts i and o refer to the inner and outer line, respectively.

For all practical applications, the thermal resistance of the vacuum space (R_2) predominates over the air/outer wall (R_1) and the inner wall/fluid (R_3) resistances. The result is that the

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133

BASIC TRANSFER LINE CROSS-SECTION

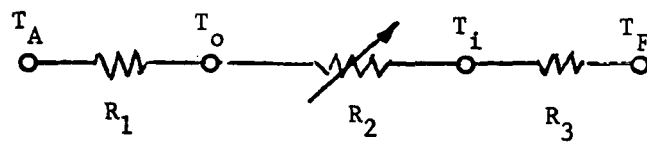
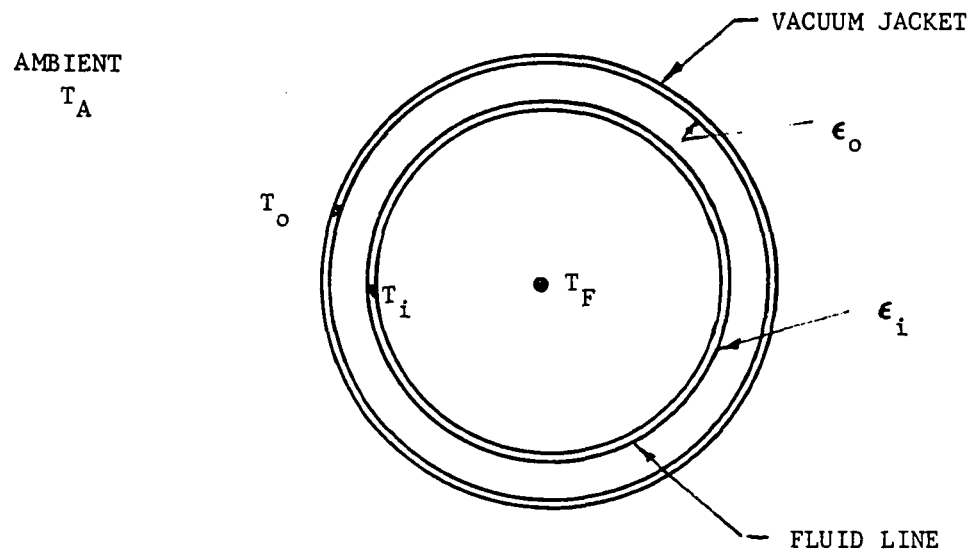


FIGURE B-1

jacket wall temperature can be approximated by the ambient temperature and the inner wall temperature by the fluid temperature.

The heat transferred from the environment to the fluid, at steady-state conditions, was evaluated for different inner line sizes and wall surface emissivities. The inner line nominal diameters were 6-in, 8-in, 10-in and 12-in; the geometric dimensions of the line system are listed in Table B-1.

Table B-1. Vacuum Jacket and Inner Line Dimensions

Nominal Inner Line Size (inches)	Outside Dia.	Wall Thickness	Outside Dia.	Wall Thickness
6	6.625	0.109	8.625	0.148
8	8.625	0.109	10.750	0.165
10	10.750	0.134	12.750	0.180
12	12.750	0.156	16.0	0.250

The surface emissivity combinations investigated were $\epsilon_i = 0.03$, $\epsilon_o = 0.30$ and $\epsilon_i = 0.03$, $\epsilon_o = 0.03$. The first combination is for a polished inner line external surface and "as received" stainless steel vacuum jacket internal surface. The second combination is for both surfaces polished to minimize the radiative heat transfer across the vacuum jacket annulus.

The heat leakage rates as functions of the vacuum jacket wall temperature are plotted in Figures B-2 and B-3 for the four line sizes. The jacket temperature range extends to 54.4°C (130°F) to include the effect of solar heating on white-painted lines. The leakage rates do not include the effect of the vacuum jacket spacers. This effect is investigated separately.

Vacuum Jacketed Line with MLI

The MLI insulated vacuum jacketed line contains multilayer insulation in the annulus between the two concentric pipes to decrease heat transfer by radiation. Theoretically, the multilayer insulation will reduce the heat transfer to a value of $1/(n + 1)$ of that of straight vacuum, where n is the number of layers that forms the insulation. However, in practice, conduction through the spacer material, insulation venting perforations, insulation joints and

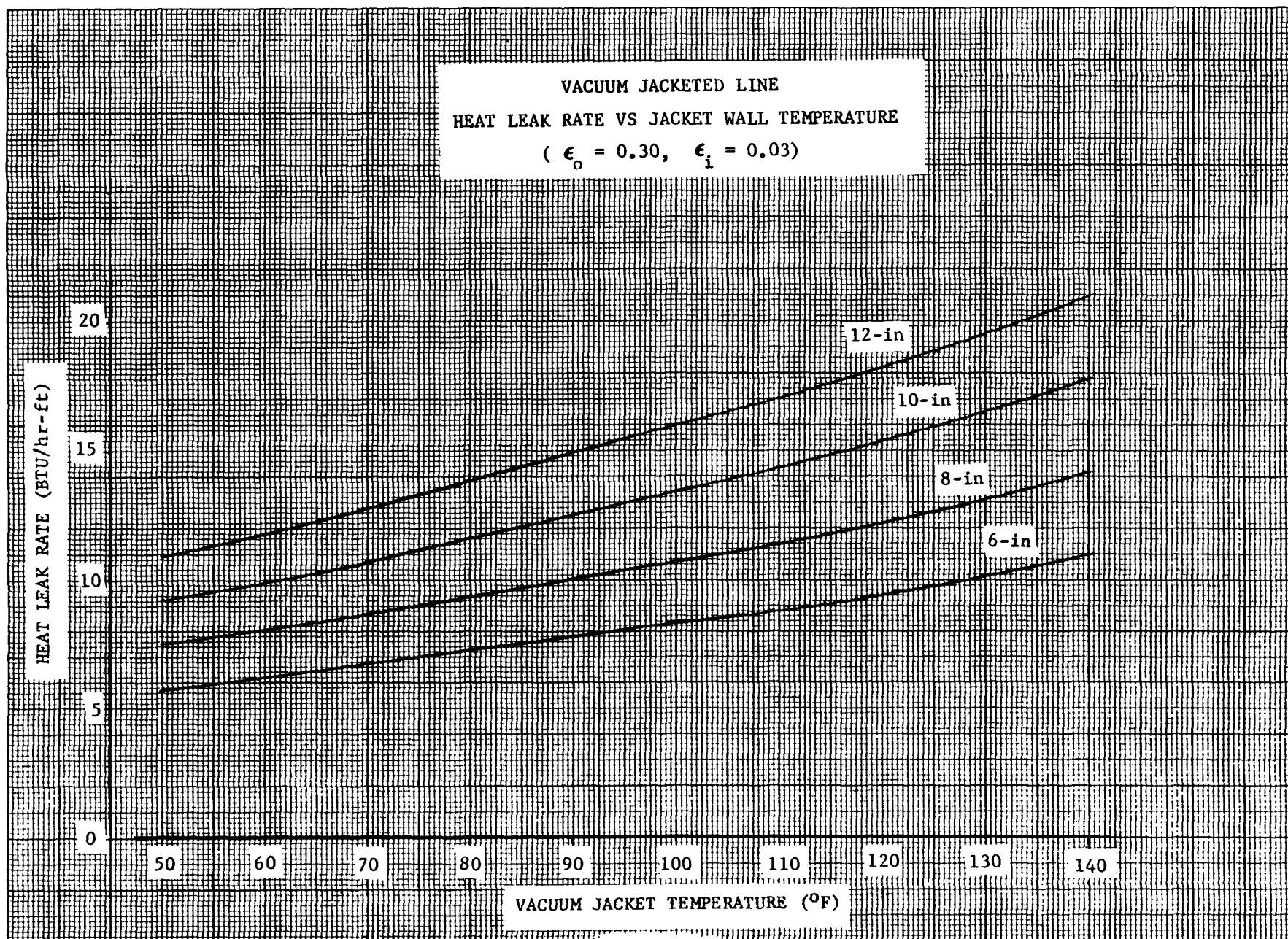


FIGURE B-2

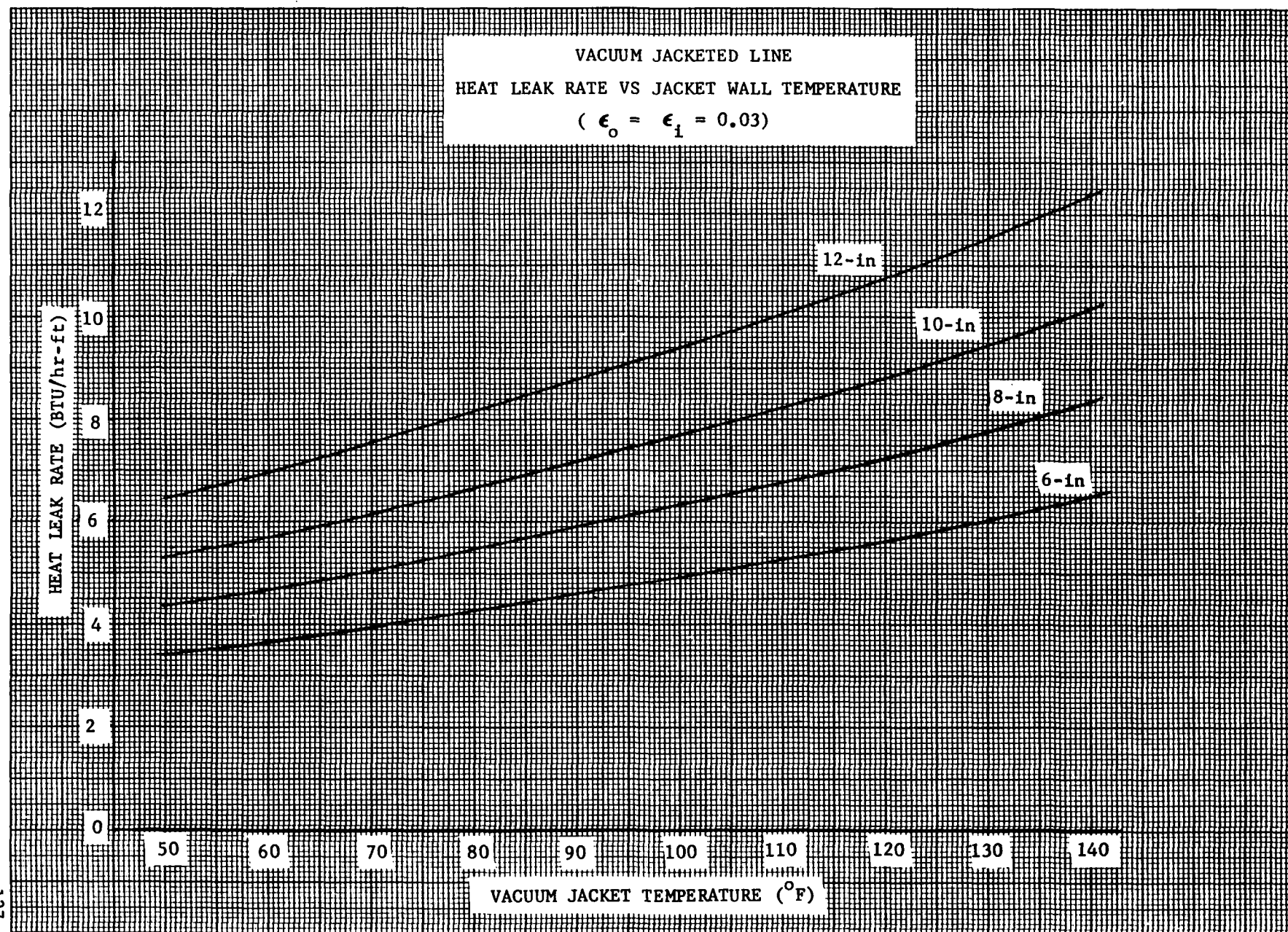


FIGURE B-3

compressive loads considerably degrade the insulation performance.

The insulation used in the vacuum jacket is composed of 20 layers of aluminized mylar, 9.4 microns (0.25 mil) thick, with dacron net used as a spacer. The heat transferred by radiation through a multilayer insulation composed of N shields is defined by the equation:

$$Q_{\text{Rad}} = \epsilon_{\text{eff}} \sigma A (T_H^4 - T_C^4) \quad (\text{B3})$$

$$\text{where } \epsilon_{\text{eff}} = \frac{1}{(N-1) (1/\epsilon_1 + 1/\epsilon_2 - 1)} \quad (\text{B4})$$

For $\epsilon_1 = 0.40$ (mylar side) and $\epsilon_2 = 0.02$ (aluminum side) the effective emittance is equal to 0.001. This is, however, the emittance of an idealized insulation; practical cases have shown emittance values considerably higher, by a factor of five or greater, depending on the complexity of the application. For the vacuum jacket insulation, a factor of 5 is selected ($\epsilon_{\text{eff}} = 0.005$), with the stipulation that the insulation is loosely wrapped around the line so that the conduction through the spacer material is reduced to a minimum.

The plots of Figure B-4 show the heat transfer for unit of length of basic line for the same inner pipe diameters as in the straight vacuum case. The heat leaks shown are for vacuum levels of 10^{-5} torr or better since they do not include gaseous conduction effects. The residual gas pressure in the vacuum annulus can contribute significantly to the line heat leakage. This contribution increases up to approximately 10^{-1} torr where it levels off.

To evaluate the heat leak at vacuum levels above 10^{-5} torr, it is assumed that at 10^{-5} torr and with the idealized insulation emittance of 0.001, the heat transferred through the insulation is by radiation only and the gas conduction is zero. In this case, for example, the 25cm (10-in) line heat leak is 0.364 w/M (0.379 Btu/hr-ft). Using the experimental thermal conductivity data at gas pressures larger than 10^{-5} torr reported in Reference 16, the heat leak increase due to residual gas conduction can be determined. The information is tabulated in Table B-2 showing the effect of higher annular space pressures on heat leak rate.

Vacuum Jacket Spacers

The spacers support the liquid line within the vacuum jacket.

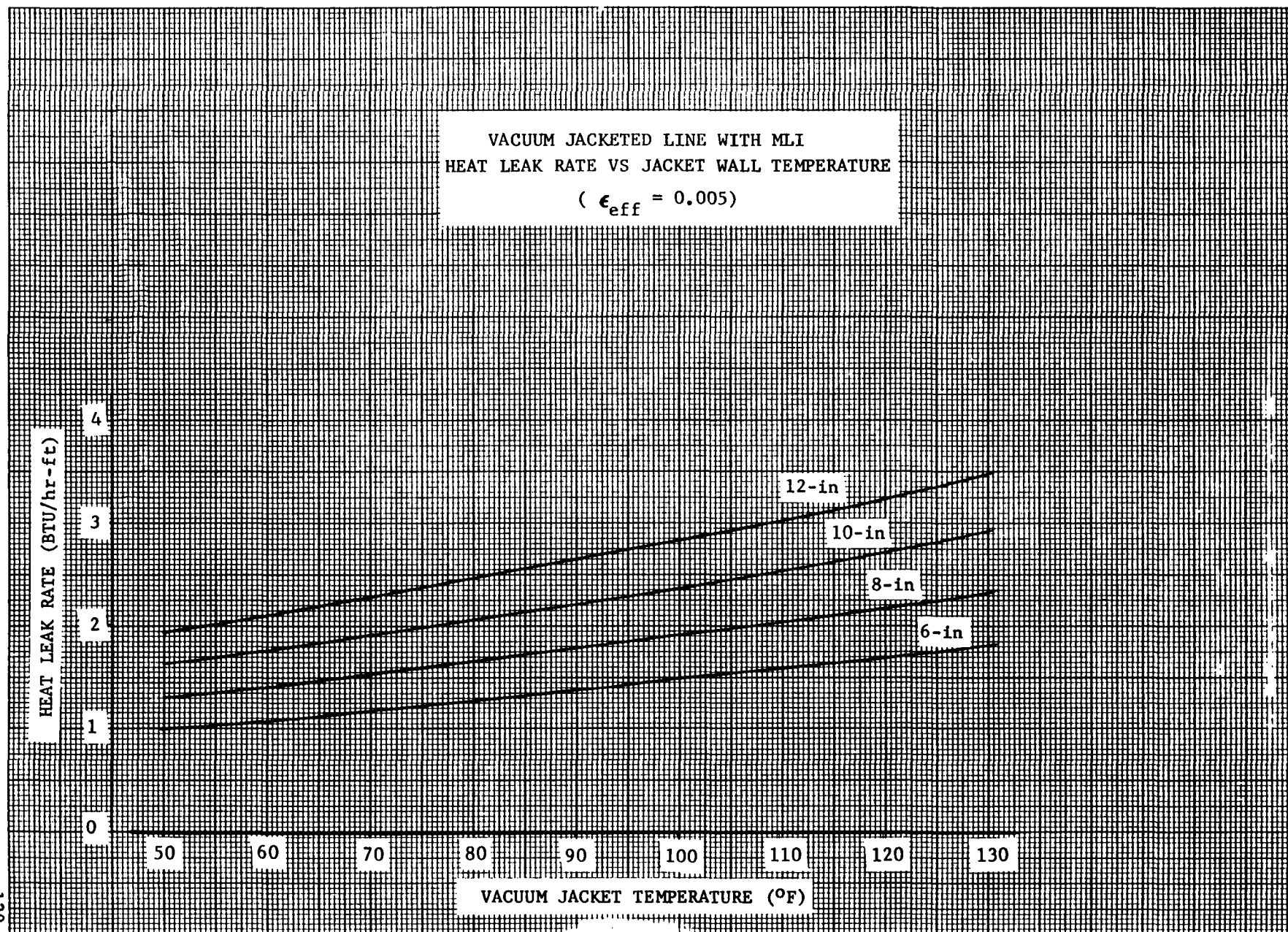


FIGURE B-4

They are designed to provide small contact area at the liquid line and at the vacuum jacket to minimize the heat transfer from the jacket to the inner pipe. The spacer used in this analysis is a four-pin assembly made of 25% glass-filled fluorocarbon plastic with a contact area of approximately 1.8 cm^2 (0.28 in^2) for the 15 cm (6-in) and 20 cm (8-in) lines and approximately 2.8 cm^2 (0.44 in^2) for the 25 cm (10-in) and 30.5 cm (12-in) lines. The corresponding heat transferred through the spacers with the jacket at ambient temperature and the inner line at 14°K (25°R) and 0.9 and 1.3 w (3.0 and 4.6 Btu/hr) per spacer. Assuming that each 12 m (40-ft) line section is supported at eight-foot intervals, the spacer contributions to the line leakage are 0.361 w per linear meter (0.375 Btu/hr per linear foot) of the 15 cm and 20 cm (6-in and 8-in) lines and 0.553 w per linear meter (0.575 Btu/hr per linear foot) of the 25 cm and 30.5 cm (10-in and 12-in) lines.

Table B-2. Gas Conduction Effect on Line Heat Leak

Vacuum (Torr)	q/q_0	Heat Leak Rate (530°R to 25°R) (Btu/hr-ft)			
		6-in	8-in	10-in	12-in
10^{-5}	1.0	0.234	0.305	0.380	0.451
10^{-4}	1.25	0.293	0.381	0.475	0.564
10^{-3}	1.875	0.439	0.572	0.712	0.846
10^{-2}	5.50	1.289	1.678	2.090	2.481
10^{-1}	31.25	7.324	9.537	11.875	14.099

Basic Line Section Heat Leak

The basic line section heat leak rate, expressed in Btu/hr per linear foot, is the average heat leak over a 12.2 m (40-ft) section including the spacers. To simplify the reporting of the results, the heat leaks are quoted only for a vacuum jacket temperature of 294°K (21°C) (530°R (70°F)) and an inner line temperature of 14°K (25°R). These heat leaks are applicable to the transfer of both slush and triple point LH_2 14°K (25°R) and triple point LOX 54°K (98°R) since the difference in radiation heat transfer is insignificant and the uncertainties in solid conduction do not warrant corrections of the quoted values.

The heat leaks for vacuum jacketed lines with and without MLI are

reported in Table B-3.

The comparison of these two insulation techniques in terms of steady-state heat leak rates shows that the leaks for the basic section of the MLI insulated line are 1/3 to 1/5 of those for the plain vacuum jacketed line depending on whether the comparison is made with the polished or "as received" inner surfaces of the vacuum jacket.

Table B-3. Basic Line Section Heat Leak Rates w/m (Btu/hr-ft) - 294°K to 14°K (530°R to 25°R)

Line Size, cm (inches)	V.J. without MLI		V.J. with MLI
	$\epsilon_j = 0.30, \epsilon_L = 0.03$	$\epsilon_j = \epsilon_L = 0.03$	$\epsilon_{eff} = 0.005$
15 (6)	7.03	4.34	1.55
20 (8)	9.02	5.45	1.90
25 (10)	11.32	6.76	2.48
30 (12)	13.37	8.10	2.83

Final selection of MLI versus non-MLI insulation, however, should be made on the basis of economics. Economic factors to be considered include the cost of material and installation of MLI versus non-MLI line and the cost to cool down the MLI as a function of transfer duration. Since this analysis has shown a steady-state thermal advantage in favor of MLI and since the existing transfer lines at KSC use MLI, it will be assumed as baseline for this study.

Active Cooling of Vacuum Jacket

The heat transferred by radiation between the vacuum jacket and the inner lines is a function of the fourth power of the absolute temperature of the jacket wall. Therefore, a decrease of this temperature drastically reduces the heat radiated to the inner line.

Figure B-5 shows the ratio of the heat transferred at vacuum jacket temperatures lower than 294°K (530°R) to that transferred at 294°K (530°R). This ratio applies to the radiation heat transfer only. From inspection of the figure, it can be seen that if the jacket wall is maintained at 139°K (250°R) the heat transferred through the annulus is only 5% of that without the active cooling.

EFFECT OF WALL TEMPERATURE
ON
RADIANT HEAT TRANSFER

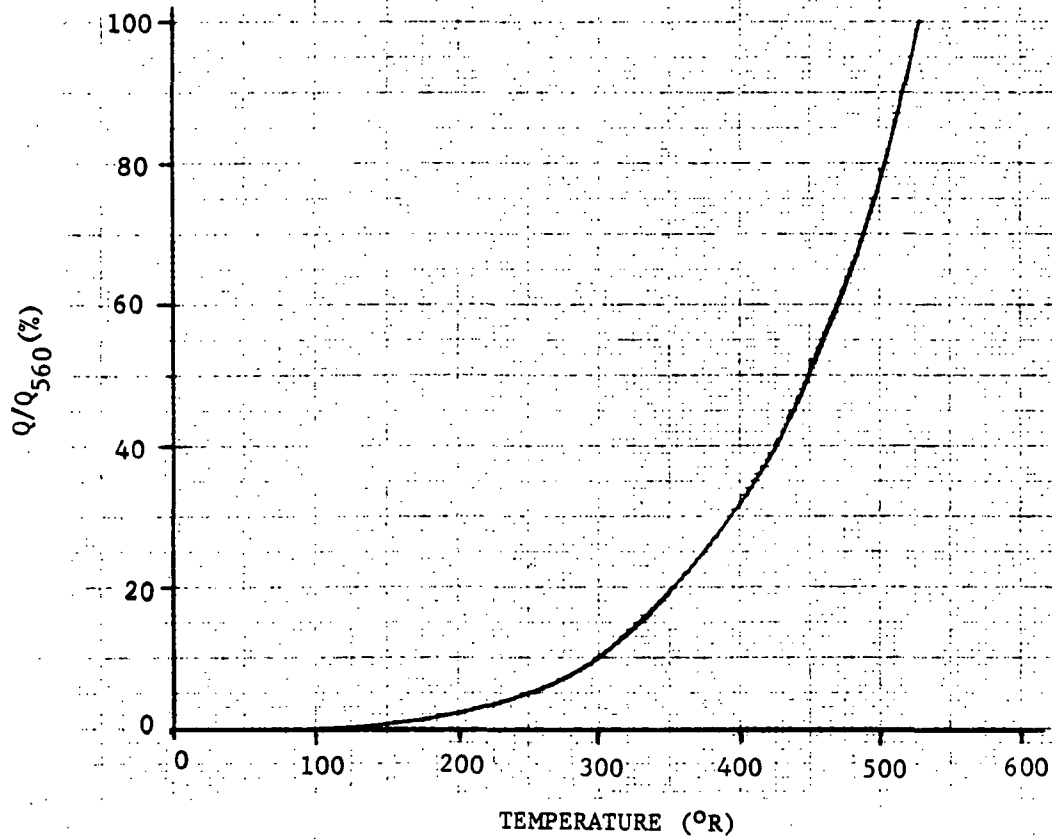


FIGURE B-5

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To evaluate the economic feasibility of the concept, active cooling was applied on the 25 cm (10-in) line used for the transfer of the SLH₂. For the analysis, it was assumed that nitrogen in liquid and vapor phases, would flow through an annulus formed by the vacuum jacket and a surrounding pipe covered with conventional insulation.

The coolant quantity requirements include that required to chill-down the vacuum jacket wall, the external insulation and pipe, and that required to remove the environmental heat load. Since the vacuum jacket chilldown requirement is the largest, it was decided to determine the transfer duration whereby the cost of cooldown of the vacuum jacket would be equal to the cost of loss of solid LH₂ without active cooling.

For the analysis, it was assumed that the vacuum jacket had to be cooled down to an average temperature of 139°K (250°R) with nitrogen entering the cooling annulus as a liquid at 77.2°K (139°R) and leaving as a vapor at 194.4°K (350°R). The environmental heat leak to the SLH₂ line in the case without active cooling was taken equal to 9.6 w per meter of line (10 Btu/hr per foot of line) (see following section). In addition, it was assumed that the active cooling of the jacket would completely eliminate any heat transfer to the inner line.

The weight of a linear meter (foot) of vacuum jacket for the 25 cm (10-in) line [OD = 32.28 cm (12.75 in), wall thickness = 0.457 cm (0.180 in)] is 36.1 kg (24.3 lb). With an average Cp = 0.402 kJ/kg - °K (0.096 Btu/lb-°R) (Reference 30) the heat to be removed to cooldown the jacket is

$$Q_{c_j} = 36.1 \times 0.402 \times (294 - 139) = 2250 \text{ kJ/m}$$

$$(Q_{c_j} = 24.3 \times 0.096 \times (530 - 250) = 653 \text{ Btu/ft})$$

The amount of LN₂ required for this cooldown is

$$W_{LN_2} = \frac{Q_{c_j}}{\Delta H} = \frac{2250}{323} = 7.0 \text{ kg per meter}$$

$$(W_{LN_2} = \frac{Q_{c_j}}{\Delta H} = \frac{653}{139} = 4.7 \text{ lb per foot}) \quad (B5)$$

where ΔH is the fluid enthalpy change. Since the cost of LN_2 is \$66.04/m³ (\$0.25 per gallon), the cost of jacket cooldown is

$$\frac{7.0}{806} \times 66.04 = \$0.574 \text{ per meter}$$

$$\left(\frac{4.7}{6.73} \times 0.25 = \$0.175 \text{ per foot}\right).$$

For the uncooled jacket case, the solid H_2 melted during the duration t in hours is:

$$W_{\text{SOLID}} = \frac{\dot{Q}_{\text{LINE}}}{H_{\text{FUSION}}} \times t = \frac{9.62}{16.2} \times t = \left(\frac{10.0}{25.1} \times t\right) \quad (\text{B6})$$

where \dot{Q}_{LINE} is the heat leakage rate through the vacuum jacket and H_{FUSION} is the hydrogen heat of fusion.

Since the cost of producing solid from triple point hydrogen is approximately \$0.71/kg (\$0.32/lb) (see Section II.C), the cost of melted solid during the transfer duration t is:

$$\frac{9.62}{16.2} \times t \times 0.71 = \$0.574 \times t \text{ per meter}$$

$$\left(\frac{10.0}{25.1} \times t \times 0.32 = \$0.127 \times t \text{ per foot}\right).$$

Equating the cost of the LN_2 to the cost of the melted hydrogen, the duration t can be determined.

$$t = \frac{0.574}{0.416} = \left(\frac{0.175}{0.127}\right) = 1.38 \text{ hr.}$$

The transfer duration just determined is a minimum since the LH_2 required to cooldown the external insulation and line to remove the environmental heat load was not included in the LN_2 weight and cost.

The result obtained with the analysis indicated that up to transfer durations of at least 1 hour and 20 minutes SLH_2 transfer with cooled vacuum jacket costs more than with uncooled jacket. Since the planned SLH_2 loading operation is about the same duration (90 minutes), the use of active cooling is not justified.

The conclusion just presented was reached without any consideration given to the cost of design, fabrication and maintenance of the cooled line. If these elements are included (even if it would be possible to use a coolant "free" LH_2 from vehicle tank vent) the use of cooled vacuum jackets becomes even less attractive.

It should be noted that the transfer of slush and/or triple point cryogenics is feasible with the expected heat leakage rates through the uncooled vacuum jackets. Therefore, in view of the presented results, no further consideration will be given to the vacuum jacket active cooling.

Comparison with Manufactured Data

The basic piping heat leak rates shown in Table B-3 and the joint heat leaks previously reported are the minimum that can be expected at steady-state from an insulated line configuration under favorable vacuum conditions. However, other factors that cannot completely be accounted for analytically degrade the system performance. For example, the actual installation of the multilayer insulation in the vacuum annulus and around the welded joint and the expected vacuum level in the annulus all impact the system performance. It is the purpose of this section to compare the analytically-derived heat leak rates with the data quoted by the manufacturer of most of the transfer system installed at KSC and to derive a factor that compensates for the differences.

The Linde Company Design Manual for vacuum insulated piping (Reference 16) defines the heat leak rates for 15 cm (6-in) and 20 cm (8-in) line and joints listed in Table B-4. The values for the 25 cm (10-in) and 30.5 cm (12-in) sizes were extrapolated from the other line size values. Table B-4 also presents the comparison of the Linde heat leak rates with those analytically derived.

The comparison shows that the Linde values are up to 20% higher than those calculated. Therefore, considering the analysis uncertainties in assessing the insulation performance and in simulating the welded joint, the Linde data will be used in place of the analytical data to allow a more conservative system analysis.

KSC Transfer System Heat Leak Rates

The KSC LH_2 and LOX transfer systems are composed of components listed in Table B-5. The table also lists the heat leakage rates of each component. These leakages were determined from the Linde Design Manual or based on information contained in Reference 36.

Table B-4. Comparison of Line and Joint Heat Leak Rates

Heat Leak Source	Line Size, cm (inches)			
	15 (6)	20 (8)	25 (10)	30.5 (12)
Line w/m (Btu/hr-ft)				
Linde	1.78 (1.85)	2.28 (2.37)	2.69 (2.80)	3.27 (3.40)
Analysis	1.54 (1.60)	1.89 (1.97)	2.47 (2.57)	2.83 (2.94)
Δ /Analysis	16%	20%	9%	16%
Joint w (Btu/hr)				
Linde	7.0 (24.0)	8.7 (29.6)	9.8 (33.6)	11.9 (40.5)
Analysis	6.0 (20.3)	7.6 (25.8)	9.2 (31.5)	11.2 (38.2)
Δ /Analysis	18%	15%	7%	6%

For this study, it is convenient to average the total heat leakage rate over the total length of the transfer system and to define a leakage per linear m(ft) of line. This quantity is a characteristic of the system and can be used without introducing large errors even if the system is slightly different.

The total leakage rates of Table B-5 averaged over the total system lengths give a leakage rate of 4.7 and 3.4 x per linear meter (4.9 and 3.5 Btu/hr per linear foot) for the LH₂ and LOX lines, respectively. These leakage rates apply to stabilized transfer conditions. i.e., when the insulation has reached steady-state conditions.

The Linde Design Manual presents the stabilization time as a function of steady-state heat transfer rate for vacuum insulated lines. The application of this information, per Linde Design Manual procedure, to the transfer lines results in the heat leak rates listed in Table B-6.

Regardless of the analyses and manufacturer data available, uncertainties still exist as to the actual leakage rates of the two transfer systems. Therefore, for this analysis it was decided that the use of time-varying rates was not justified and that nominal heat transfer rates of 9.6 w/m (10 Btu/hr-ft) and 5.8 w/m (6 Btu/hr-ft) for the existing LC 39 LH₂ and LOX lines, respectively, would be used. For varying sizes of MLI-vacuum jacketed line, therefore, a heat leak value of 0.38 w/cm dia/m of line (1 Btu/hr/inch dia/ft of line) will be used.

Table B-5. KSC Transfer System Components and Heat Leakage Rates

Component	Unit Leakage Rate	Total Leakage Rate w (Btu/hr)
<u>LH₂ System, 25 cm (10-in Line)</u>		
Insulated Line Sections, 533.4 m (1750 ft)	2.7 w/m (2.8 Btu/hr-ft)	1436 (4900)
Welded Joints (56)	10.0 w (34.0 Btu/hr)	558 (1904)
Mechanical Joints (6)	17.6 w (60.0 Btu/hr)	106 (360)
Elbows (12)	2.5 w (8.5 Btu/hr)	30 (102)
Tees (3)	3.8 w (13.0 Btu/hr)	11 (39)
Valves (4)	87.9 w (300.0 Btu/hr)	352 (1200)
Total		2493 (8505)
<u>LOX System, 15 cm (6-in Line)</u>		
Insulated Line Sections 506.6 m (1662 ft)	1.78 w/m (1.85 Btu/hr-ft)	901 (3075)
Welded Joints (60)	7.0 w (24 Btu/hr)	422 (1440)
Elbows (11)	1.8 w (6 Btu/hr)	19 (66)
Tees (7)	1.5 w (5 Btu/hr)	10 (35)
Reducers (3)	4.1 w (14 Btu/hr)	12 (42)
(20 cm 8-in Line)		
Insulated Line Sections 32.0 m (105 ft)	2.28 w/m (2.37 Btu/hr-ft)	73 (249)
Welded Joints (12)	8.7 w (29.6 Btu/hr)	104 (355)
Elbows (8)	2.2 w (7.4 Btu/hr)	17 (59)
Valves (3)	88 w (300 Btu/hr)	264 (900)
Total		1823 (6221)

Table B-6. LH_2 and LOX Line Heat Leak Rates

Time from Start of Insulation Cooldown (min)	Q/Q Steady State	Heat Leak Rate, w/m (Btu/hr-ft)	
		LH_2 25 cm dia (10-in dia)	LOX 15 cm dia (6-in dia)
10	4.1	19.3 (20.1)	13.8 (14.3)
20	3.0	14.1 (14.7)	10.1 (10.5)
30	2.3	10.9 (11.3)	7.7 (8.05)
50	1.7	8.0 (8.3)	5.7 (5.9)
100	1.3	6.2 (6.4)	4.3 (4.5)
1000	1.1	5.2 (5.4)	3.7 (3.8)

APPENDIX C

SSTO FUEL TANK INSULATION ANALYSIS

The fuel tank for the baseline SSTO vehicle as defined in Reference 1 is an integral, multilobe, aluminum isogrid structure which conforms to the forward fuselage shape and provides the primary structural load paths. The fuselage is attached to the tank at the isogrid nodal points with 10 cm (4-in) titanium standoff supports as shown in Figure C-1. This configuration is the basis for this analysis.

Ascent stage liquid hydrogen tanks have been used both internal and external insulation for limiting boiloff and preventing air liquefaction. Internal insulation offers several advantages. These include:

1. Cooldown time is greatly reduced. Accumulation of liquid in the tank starts almost as soon as the fill line is cooled.
2. Tank support structure is simplified since it does not penetrate the insulation.
3. The insulated surface (tank interior) is accessible to installation, inspection, repair and modification of the insulation.
4. The tank shell operates at a higher temperature, resulting in smaller thermal dimensional changes.
5. Additional protection of the insulation is not required. Internal insulations are designed to withstand thermal stresses and forces due to tank pressurization and propellant motion.

Disadvantages to the internal insulations, compared to external are:

1. Higher thermal conductivity. The more attractive "gas layer" internal insulations depend on permeation of the vapor of the contained liquid to prevent stresses due to tank (and hydrostatic) pressure. Therefore, for liquid hydrogen, the thermal conductivity of gaseous hydrogen limits the effectiveness of the insulation. "Sealed" type internal insulations may exhibit a performance improvement. However, for a frequently reused system, it is likely that permeation of hydrogen into the insulation will ultimately degrade its performance.

SSTO PROPELLANT TANK AND INSULATION SYSTEM

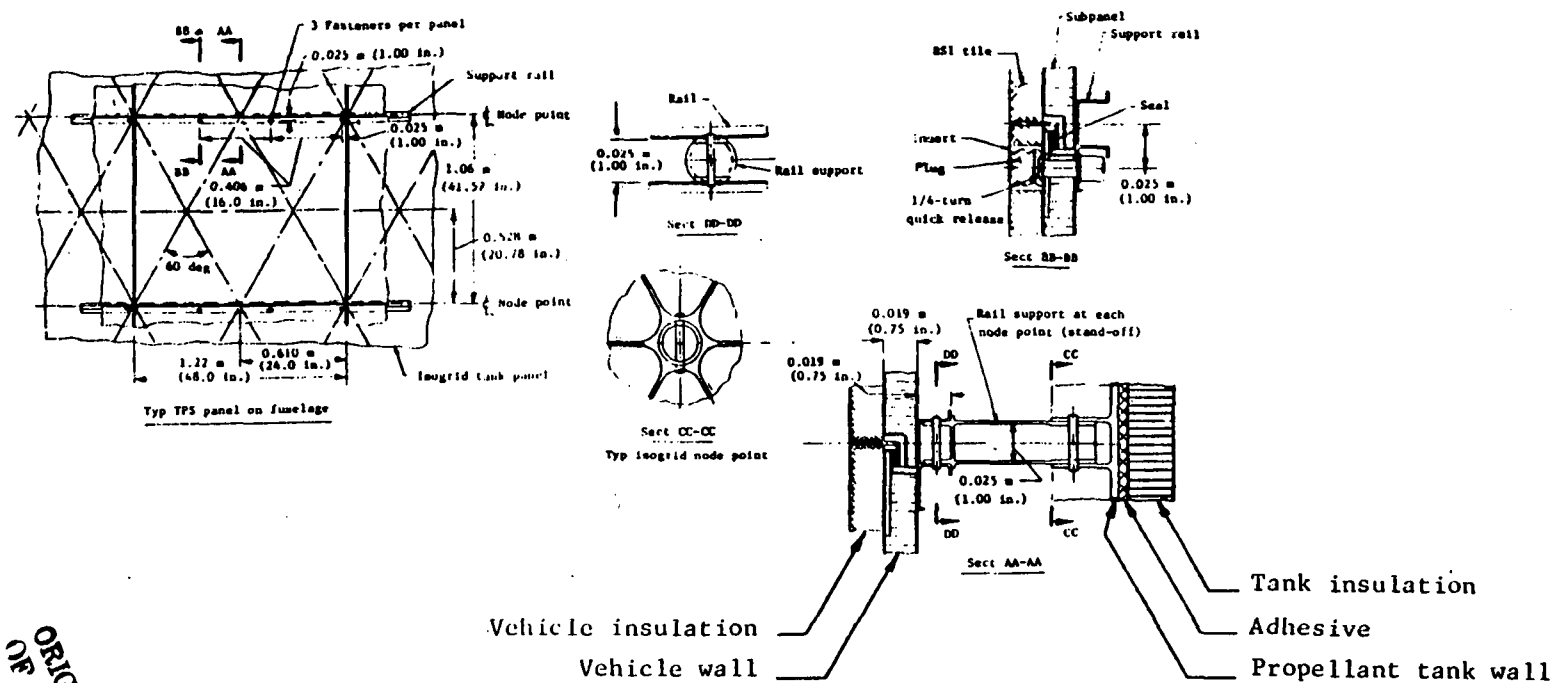


FIGURE C-1

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2. Internal insulations may be heavier than external systems, although the PPO foam system is comparable to external foam. In addition, for a given volumetric capacity, a tank with internal insulation must be larger and, therefore, heavier to accommodate the displacement of the insulation.

In view of the above, it is apparent that internal insulation should be the choice for SSTO. Therefore, the two most promising internal "gas layer" systems (PPO foam and capillary) were evaluated.

PPO foam has been developed and tested for liquid hydrogen applications by Convair Division of General Dynamics (Reference 37). Functionally this insulation resembles a bundle of tiny tubes or soda straws all oriented perpendicular to the tank wall and sealed at the wall end. By virtue of capillary forces, a stable meniscus forms in each tube at the interface between liquid and gas, thereby trapping the gas in the foam. The size of the openings is sufficiently small to eliminate convection, and heat is conducted via the contained gas and the solid foam structure.

Functioning of the PPO foam insulation is dependent on a zero or very low permeability through the tube walls. The effect of permeability is to permit an upward flow of vapor through the insulation. This results in penetration of liquid into the insulation thickness. In fact, PPO foam does exhibit a significant permeability in the direction parallel to the tank wall and its thermal conductivity is greater than that due to gas and solid conduction alone. As a result, thermal conductivity measurements for vertical insulation panels are on the order of 50% greater than those made with horizontal orientation.

The second gas layer insulation concept considered is the capillary insulation system originated and developed by Martin Marietta (Reference 38). This insulation is based on a honeycomb structure bonded to the tank wall to form discrete cells. The cells are filled with a lightweight material such as fiberglass, to inhibit convection. The cells are closed on the liquid side by a face sheet made of a thin film of a suitable nonmetal. The face sheet is perforated with one (or more) small holes for each cell. This opening permits pressure equalization between the tank interior and the insulation, eliminating pressure loading. The hole size is chosen small enough typically 0.75 to 1.50 mm (0.030 to 0.060 inch) to permit capillary forces to form a stable meniscus at the opening, once pressure is equalized, preventing interchange of liquid and gas.

Thermal stresses are eliminated by dimpling of the face sheet to provide an excess of material which prevents tension loading when the insulation is cooled. The honeycomb is formed with an "S" curve between node points for the same reason.

The capillary insulation shows a better thermal conductivity than PPO foam, based on published data. It does not differ between horizontal and vertical orientation, since the honeycomb structure positively prevents communication between cells.

In the cases of slush and triple point liquid hydrogen, the performance of both of the gas layer type insulations might be expected to degrade since a liquid-gas interface cannot exist below the saturation temperature with a subcooled liquid. In practice, however, the thermal boundary layer buildup is adequate to permit proper functioning of the insulation, except for greater insulation thickness and lower heat flux applications. Testing of these gas layer insulations has been with saturated LH₂. However, in the case of the capillary insulation, subcooled tests have been performed by rapidly increasing tank pressure. In these tests, no effect was noted for 2.5 cm (1-inch) of insulation when the pressure was raised on the order of 140 KPa (20 psi), which would result in 3 to 4°C (6 to 7°F) subcooling of the bulk liquid. For PPO foam, the net effect of subcooling at the foam surface (if solid H₂ rests on the insulation) would be to move the liquid-gas interface into the foam so that the temperature rise through the liquid equals the subcooling. This imposes a small penalty because the thermal conductivity of LH₂ below 20°K (36°R) is comparable to the overall conductivity of gaseous hydrogen. For the same circumstances with capillary insulation, a thin layer of flexible open cell foam installed over the face sheet would permit a gradient through the liquid to bring the face sheet to the saturation temperature. Further testing of both systems at subcooled temperatures to confirm these observations will be required.

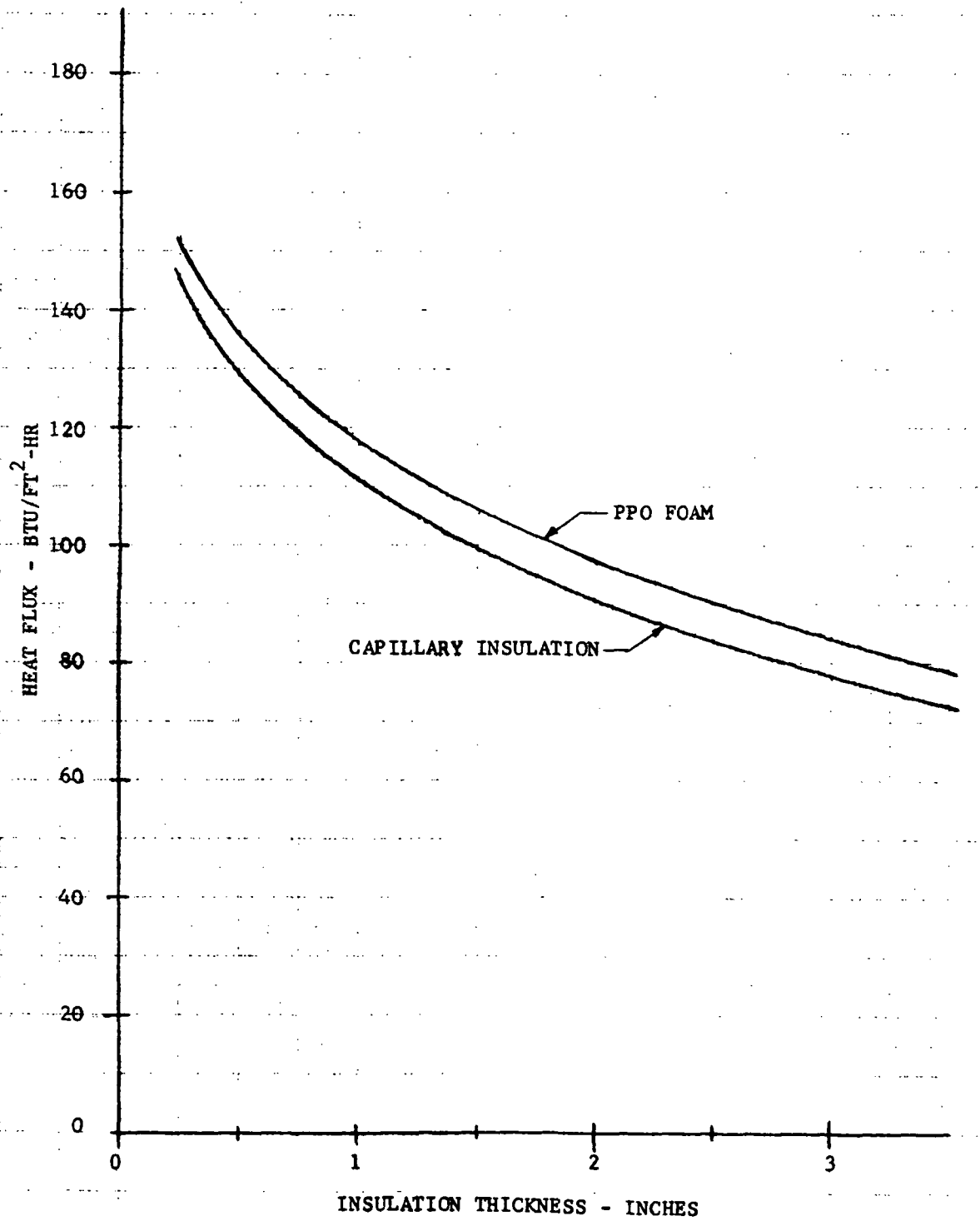
Weight of the PPO foam system is assumed to be 0.4170 kg/m² per cm of thickness (0.2169 lb/ft² per inch of thickness) plus 0.73 kg/m² (0.15 lb/ft²) for adhesive. The capillary system is considerably heavier at 0.934 kg/m² per cm of thickness (0.486 lb/ft² per inch of thickness) plus 1.17 kg/m² (0.24 lb/ft²) for adhesive and face sheet. These weights are based on Convair reports for PPO foam, and weights which have been achieved for the capillary system. Improvement in weight for capillary insulation would be expected with further development.

Using the above parameters, a trade study was conducted to determine the optimum insulation system for both 50% slush H_2 and for triple point liquid H_2 . Figure C-2 presents the steady-state heat flux vs insulation thickness and Figure C-3 shows the tank wall temperature at steady-state. When cooldown losses are also considered for a typical period of 100 minutes (nominal SSTO loading plus 45-minute pad hold), the two insulations give the same performance since the greater mass to be cooled for capillary insulation tends to offset its small performance advantage. The total heat flux for 100 minutes vs insulation thickness is shown in Figure C-4. Since triple point temperature is assumed in both cases, there is no difference between slush and triple point hydrogen.

The effect on gross liftoff weight (GLOW) of increasing fuel tank insulation thickness is shown in Figure C-5, for both the 50% SLH₂ and TP LH₂ fueled vehicles. The propellants, the size and weight of fuel and oxidizer tanks, and the vehicle weight were varied to maintain a constant mixture ratio of 6 (Reference 1) and a constant ratio of GLOW to burnout weight thereby maintaining a constant delta-V capability. A value of 49386 kg (108,878 lb) for payload and other fixed items was maintained constant for all cases (Reference 1). The non-fixed portions of the vehicle were assumed to vary directly with hydrogen tank volume, while the LH₂ tank changed as a function of $(V)^{1.5}$. Figure C-5 clearly shows the significance of insulation weight and indicates a great advantage of PPO foam over capillary insulation as it is presently conceived.

To analyze the economics of increasing the fuel tank insulation, the cost of the increase in vehicle GLOW from Figure C-5 was compared to the reduction in propellant lost due to heating from Figure C-4. The results are presented in Figure C-6 relative to the SSTO baseline vehicle with 1.65 cm (0.67-inch) PPO foam. The cost associated with a change in GLOW was taken from Figure C-7 which was plotted from data in Reference 1. The cost factor used was \$1.093/delta GLOW kg/launch (\$0.496/delta GLOW lb/launch) and is for a total program of 1710 launches in 15 years. The fuel cost used was the difference between the slush or TP cost and NBP cost as defined in Para. II.C.2. These costs are 0.31 \$/kg (0.14 \$/lb) for SLH₂ and 0.15 \$/kg (0.07 \$/lb) for TP LH₂. In Figure C-6, it is shown that the savings in propellant lost due to heating are minimal in comparison to the cost of the added weight as insulation is increased. From these results it can be concluded that factors other than optimize cost dictate insulation design. For example, from Figure C-3, an insulation thickness of 1.3 to 1.9 cm (0.5 to 0.75 inches) would prevent tank wall temperatures below the condensation point of oxygen [90.5°K (163°R)]. Allowing for uncertainties in this analysis and a small design margin, an insulation of 2.5 cm (1 inch) PPO foam is recommended.

SSTO SLH₂ AND T.P. LH₂ TANK INSULATION
HEAT FLUX VERSUS THICKNESS



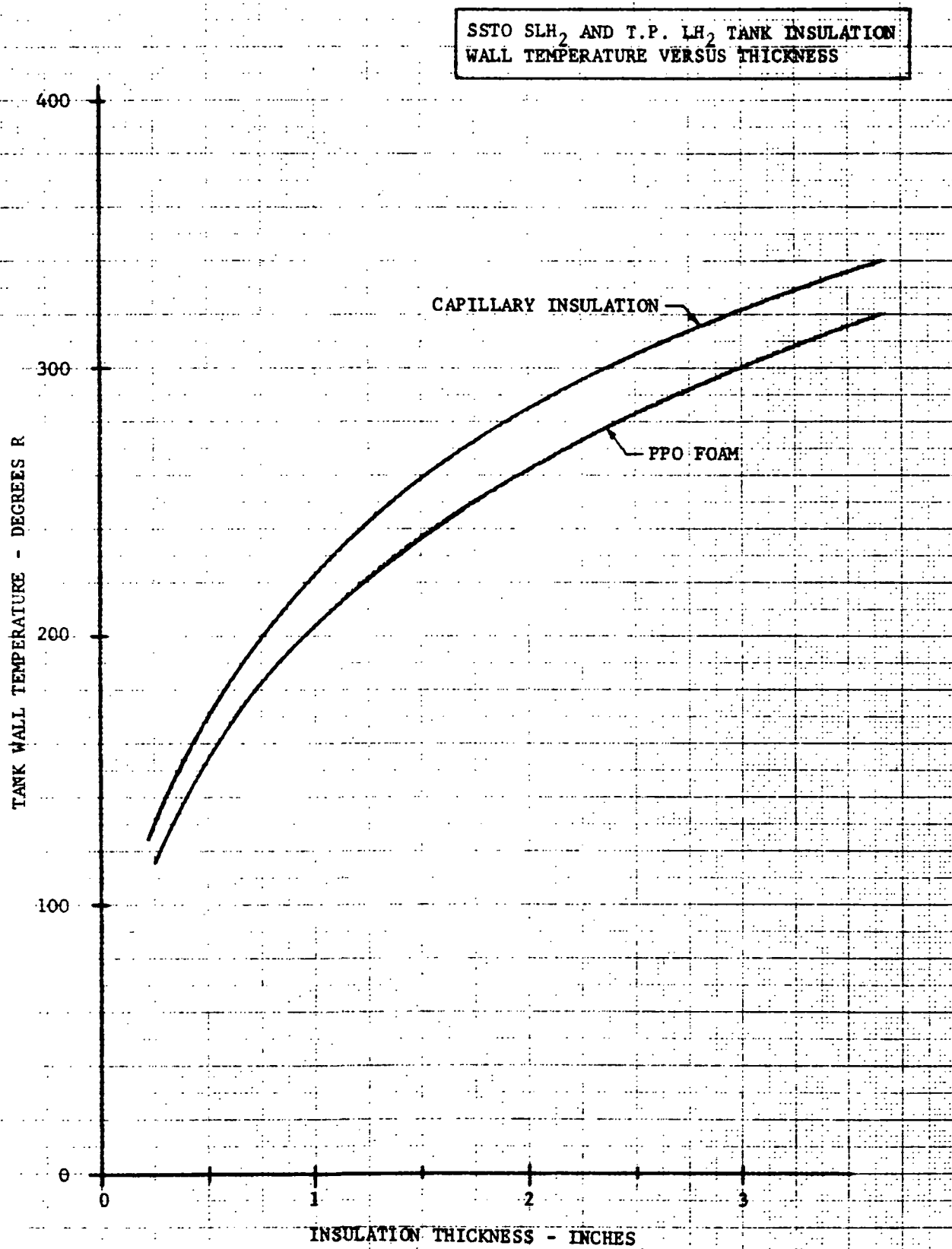
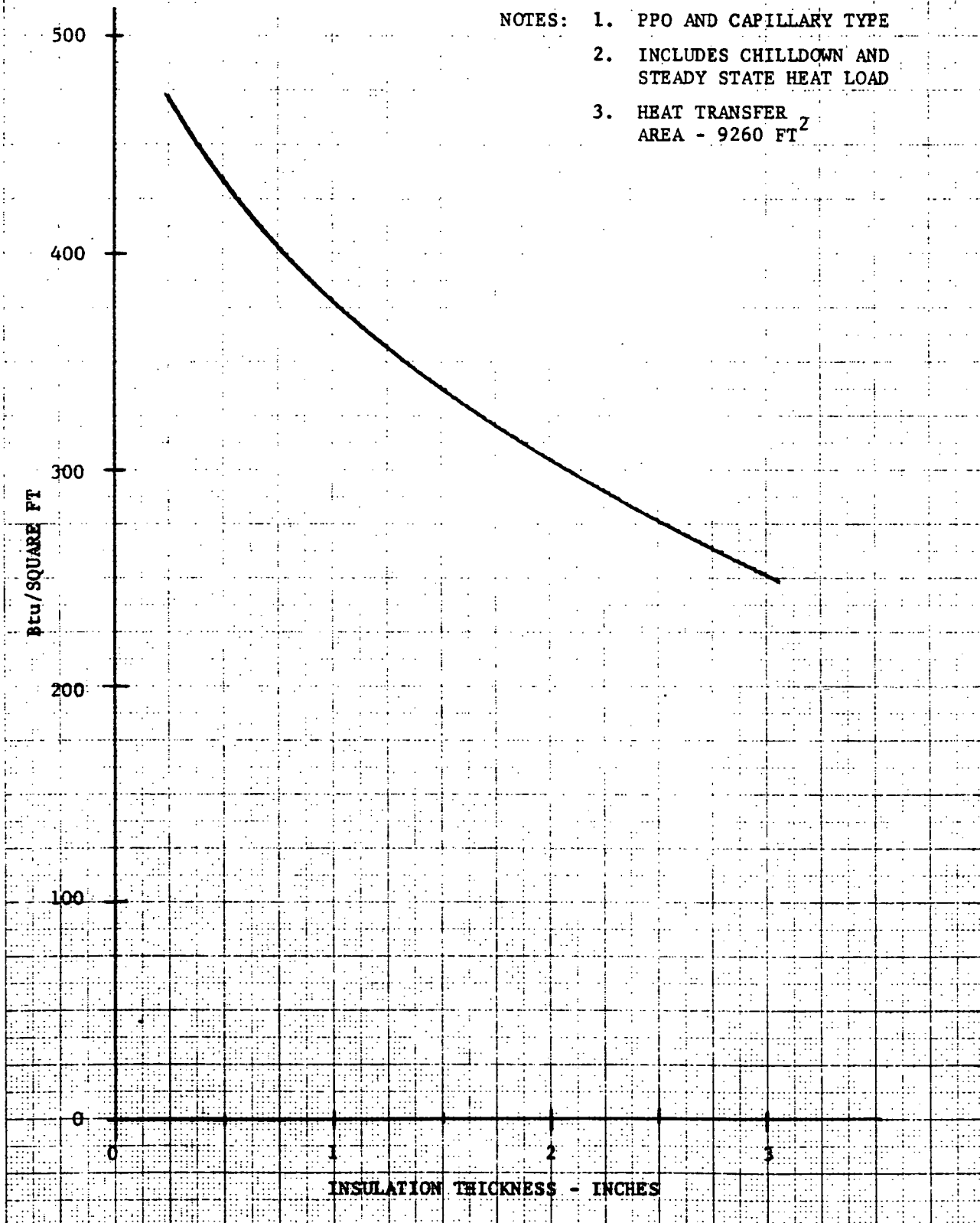


FIGURE C-3

SSTO SLH₂ AND TP LH₂ TANK INSULATION
TOTAL HEAT INPUT VERSUS THICKNESS
AFTER 100 MINUTES

- NOTES: 1. PPO AND CAPILLARY TYPE
2. INCLUDES CHILLDOWN AND
STEADY STATE HEAT LOAD
3. HEAT TRANSFER
AREA - 9260 FT²



SSTO SLH₂ AND T.P. LH₂ TANK INSULATION
GROSS LIFTOFF WEIGHT VERSUS THICKNESS

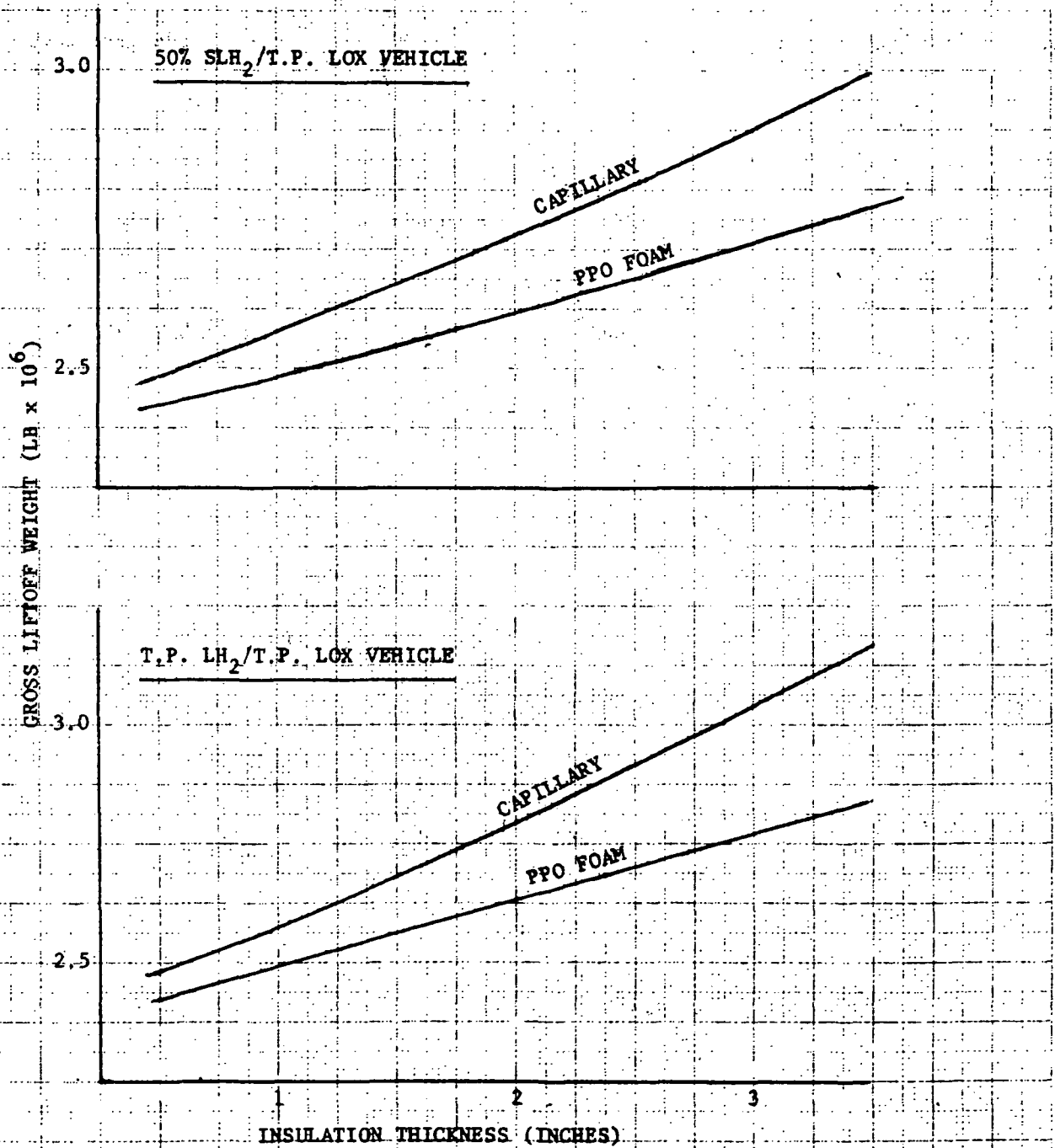
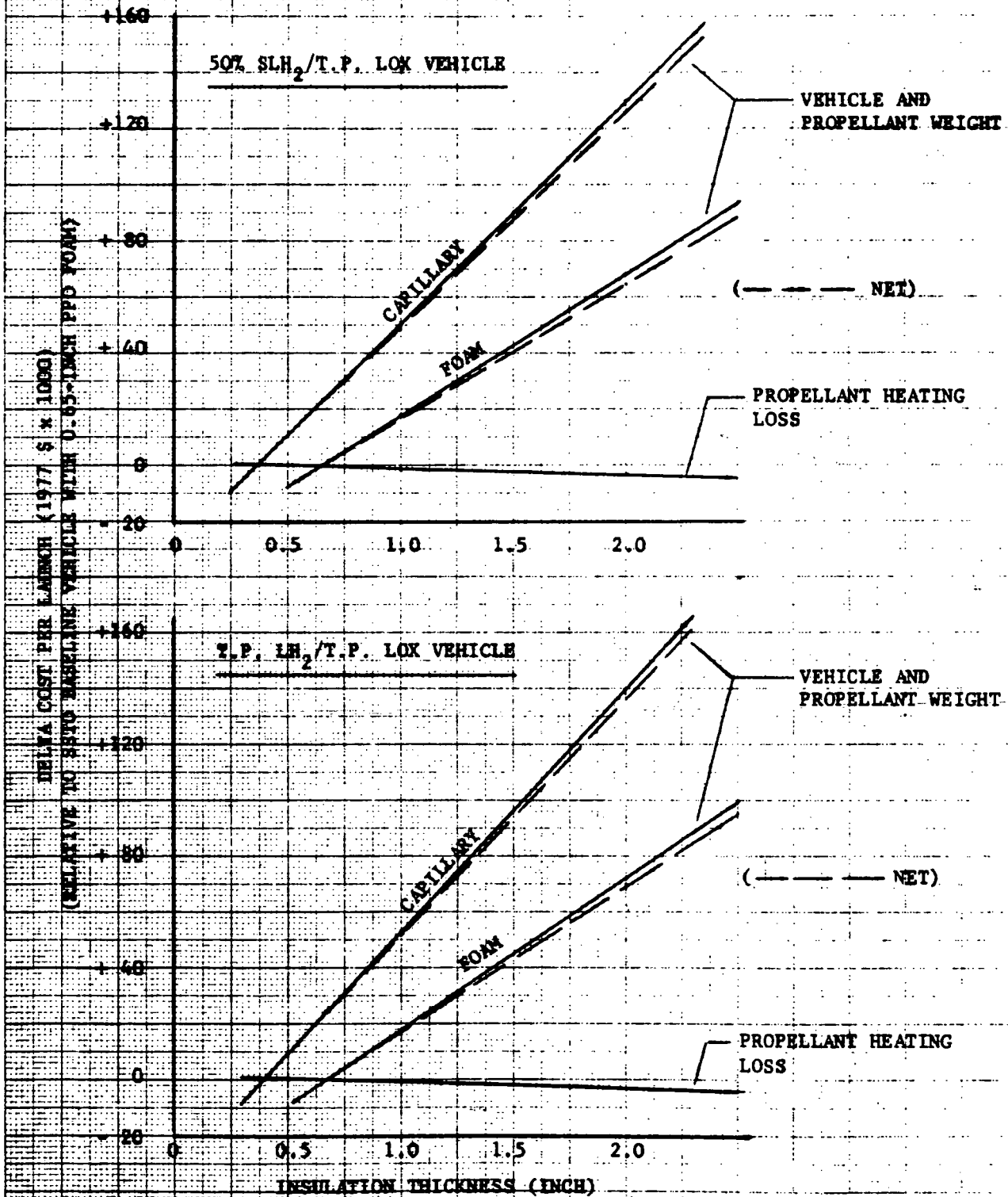


FIGURE C-5

SSTO SLH_2 AND T.P. LH_2 TANK INSULATION THICKNESS COST ANALYSIS



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SSTO TOTAL PROGRAM AND OPERATIONS
COSTS VERSUS GROSS LIFTOFF WEIGHT

NOTES:

1. PROGRAM COST INCLUDES DDTE,
PRODUCTION AND OPERATIONS.
2. SSTO LIFE CYCLE - 1710 LAUNCHES
3. PROGRAM DURATION - 15 YEARS

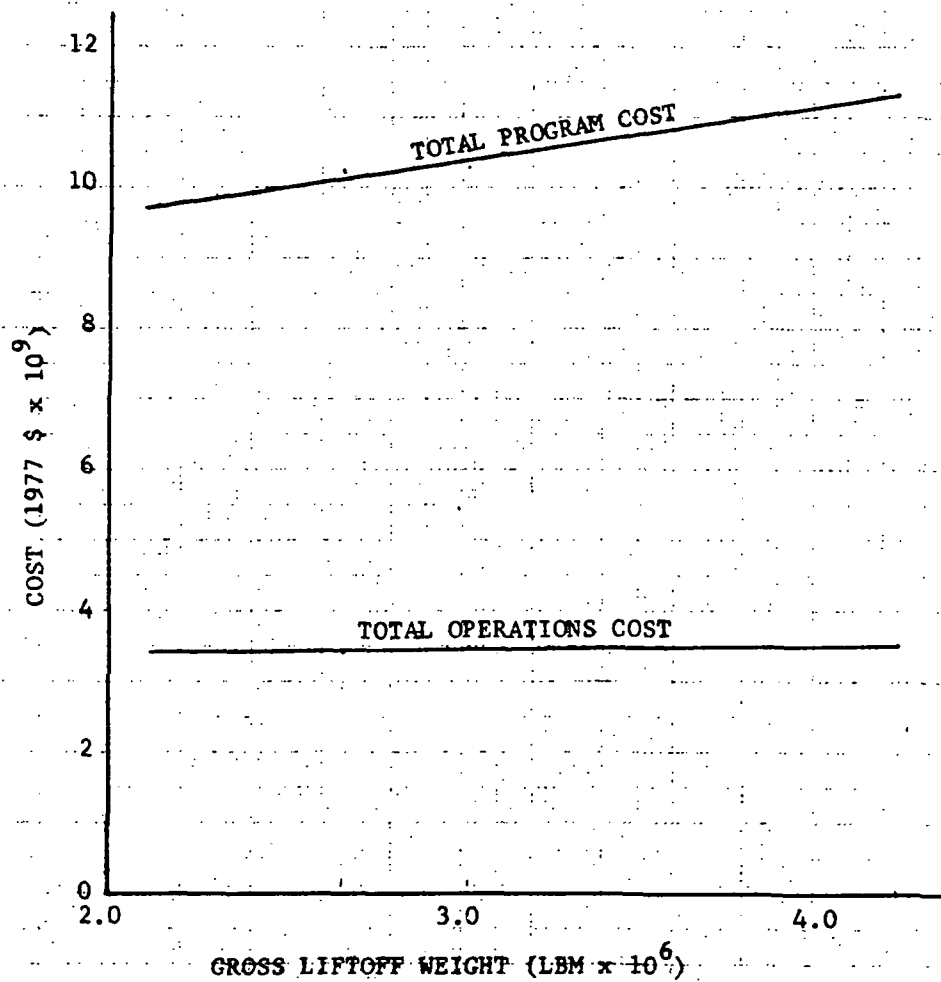


FIGURE C-7

The capillary insulation is considered to be considerably more durable and this factor, plus improvement in installed weight, could lead to its use in view of the very large number of reuses of the vehicle.

APPENDIX D

SSTO OXIDIZER TANK INSULATION ANALYSIS

LOX tankage for the baseline SSTO vehicle differs from the more typical ascent stage LOX tank in two principal ways. The tanks do not form a part of the vehicle exterior but are housed within the vehicle. Also, the triple point LOX temperature is significantly lower than the normal boiling point LOX used to date. Both of these factors are significant in determining whether LOX tank insulation is required and, if so, how much.

Insulation has generally not been required for the LOX tanks of previous boost stages. Because the NBP LOX is at or near its saturation temperature, there is no dependency of density, and therefore loaded propellant, on the rate of heat gain. For large tanks, the rate at which replenish liquid must be supplied to uninsulated tanks has been acceptable. The potential problem of a severe weight penalty due to condensation and freezing of water vapor from the air did not materialize because of the nature of frost (rather than ice) formation on extremely cold surfaces.

Triple point LOX, unlike NBP LOX, is significantly below the condensation temperature of air (or nitrogen). Therefore, the question of excessive heat transfer due to condensation on the tank exterior surface must be addressed. Further, the rate of heat gain by the lower temperature LOX will affect the replenish (pad hold) rate required to maintain the desired density. Or, for a fixed pad hold flow, the rate of heat gain will influence the density, and therefore, the quantity of LOX in the tanks. Because the LOX tanks are internally mounted, the external vehicle surfaces near the tanks will be cold, but nowhere near the temperature of an uninsulated tank. Therefore, the possibility of a different mechanism for the condensation and solidification of water vapor must be considered.

Analyses have been conducted to determine whether air will condense on an uninsulated triple point LOX tank, and if so, what rates of heat transfer are possible. For this determination, the most important factor is the convective heat transfer coefficient between the tank wall and the bulk liquid in the tank. Depending on the choice of published correlations this coefficient is found to be in the range of 227 to 1135 $\text{w/m}^2\text{-}^\circ\text{K}$ (50 to 200 $\text{Btu/hr-ft}^2\text{-}^\circ\text{R}$). For instance, the following correlation from Reference 39 gives a value for h of 775.7 $\text{w/m}^2\text{-}^\circ\text{K}$ (136.7 $\text{Btu/hr-ft}^2\text{-}^\circ\text{R}$).

$$h = 0.13 k \left[\frac{\rho^2 \beta C g_c \Delta T}{\mu k} \right]^{1/3} \quad (D-1)$$

where h = convective heat transfer coefficient

ρ = liquid density, (lb/ft³)

β = coefficient of volumetric expansion, (1/°R)

C = specific heat of liquid

g_c = gravitational constant, (ft/sec²)

T = temperature difference - bulk liquid to tank wall, (°R)

μ = viscosity, (lb/ft-sec)

k = thermal conductivity, (BTU/hr-ft-°R)

If air (or probably nitrogen if the vehicle is purged) condenses on the tank wall then a boundary condition of 78°K (140°R) (assuming nitrogen) is established. If the bulk liquid is 55°K (100°R) and the tank wall is thin (and of high conductivity such as aluminum), then a heat flux of up to 15762 w/m² (5000 Btu/hr·ft²) could result. Boiling of the LOX, which would increase the thermal resistance, is not possible since the outside boundary temperature is too low. The actual heating rate would also be dependent on the thickness of the liquid nitrogen film on the outside of the tank, the actual tank material and wall thickness, and other factors. However, it appears to be certain that condensation will occur and that it will result in a very high rate of heat gain by the LOX. Therefore, it is necessary to determine what amount of insulation is necessary and/or desirable.

A number of foam insulations are available for spray applications to the tank wall. These foam insulations, either polyurethane or isocyanurate, vary in density from 32 to 64 kg/m³ (2 to 4 lb/ft³) and in thermal conductivity from about 0.02 to 0.029 w/m·°C (0.01 to 0.017 Btu/hr·ft·°F) over the operating temperature of interest. For our analysis we assumed a typical thermal conductivity of 0.024 w/m·°C (0.014 Btu/hr·ft·°F).

The air space between the tank surface and the vehicle skin becomes an important part of the thermal resistance between the external environment and the contained LOX, so long as sufficient insulation is used to prevent air or nitrogen condensation. A combined radiation and convection heat transfer coefficient of 0.5 was assumed, although this will vary depending on separation distance (which varies around the tank). The radiation and convection co-

efficient from the vehicle exterior to the surroundings was 1.5 and ambient temperature was assumed to be 294°K (530°R). Results of the analysis of effect of insulation thickness on heat transfer rate are shown in Figure D-1. A sharp rise in heat transfer occurs as the insulation thickness goes below approximately 1.3 mm (0.05 inch). This is the point at which condensation begins to occur where the tank wall is nearest the vehicle skin. On the opposite side, where the two tanks look at each other, condensation probably occurs at much greater insulation thicknesses, but with little effect on heat transfer.

Figure D-2 shows the total heat gain of the LOX for both tanks as influenced by insulation thickness. A total of about 75,500 kJ (716,000 Btu) is given by the tank walls and approximately 3 to 10% of that value is contributed by the insulation for 6.3 to 2.5 cm (¼ to 1 inch) thickness. An estimated curve for LOX heat gain for uninsulated tanks is also shown.

These results show that after the minimum insulation is applied to prevent condensation of air or nitrogen, little is gained by increasing the insulation thickness. It is estimated that the minimum practical thickness of spray foam insulation is on the order of 1.2 cm (½ inch), and, therefore, this amount is recommended. The question of ice formation was not evaluated in great detail. However, a cursory analysis indicates that with 1.2 cm (½ inch) of foam insulation, the ice buildup rate could not exceed about 0.5 mm (0.020 inch) per hour, and would not begin until the vehicle skin cooled to below the freezing point. Additional analysis and testing on ice formation is recommended.

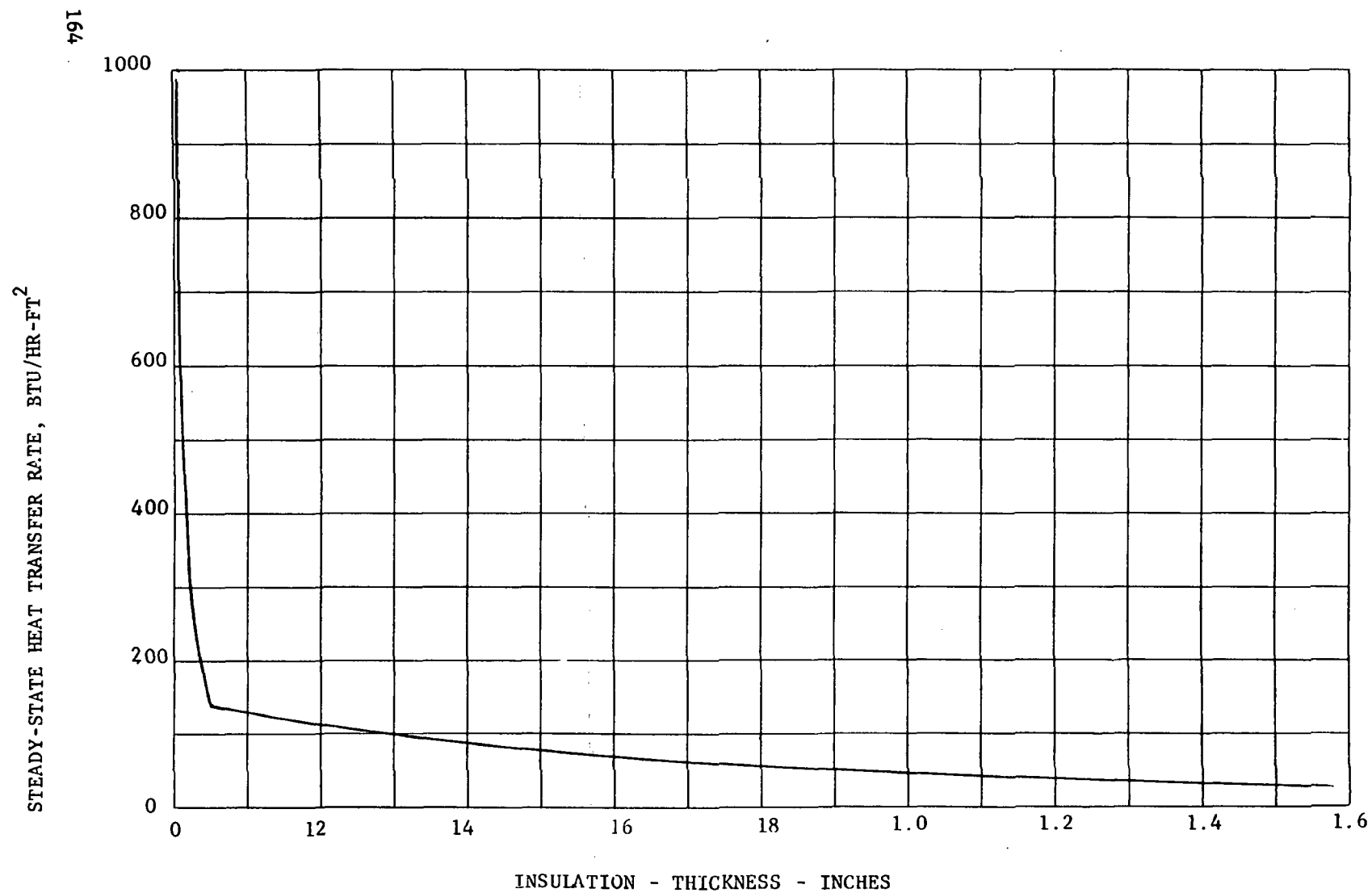


FIGURE D-1 HEAT TRANSFER TO TRIPLE POINT LOX VS. THICKNESS OF TYPICAL SPRAY FOAM INSULATION

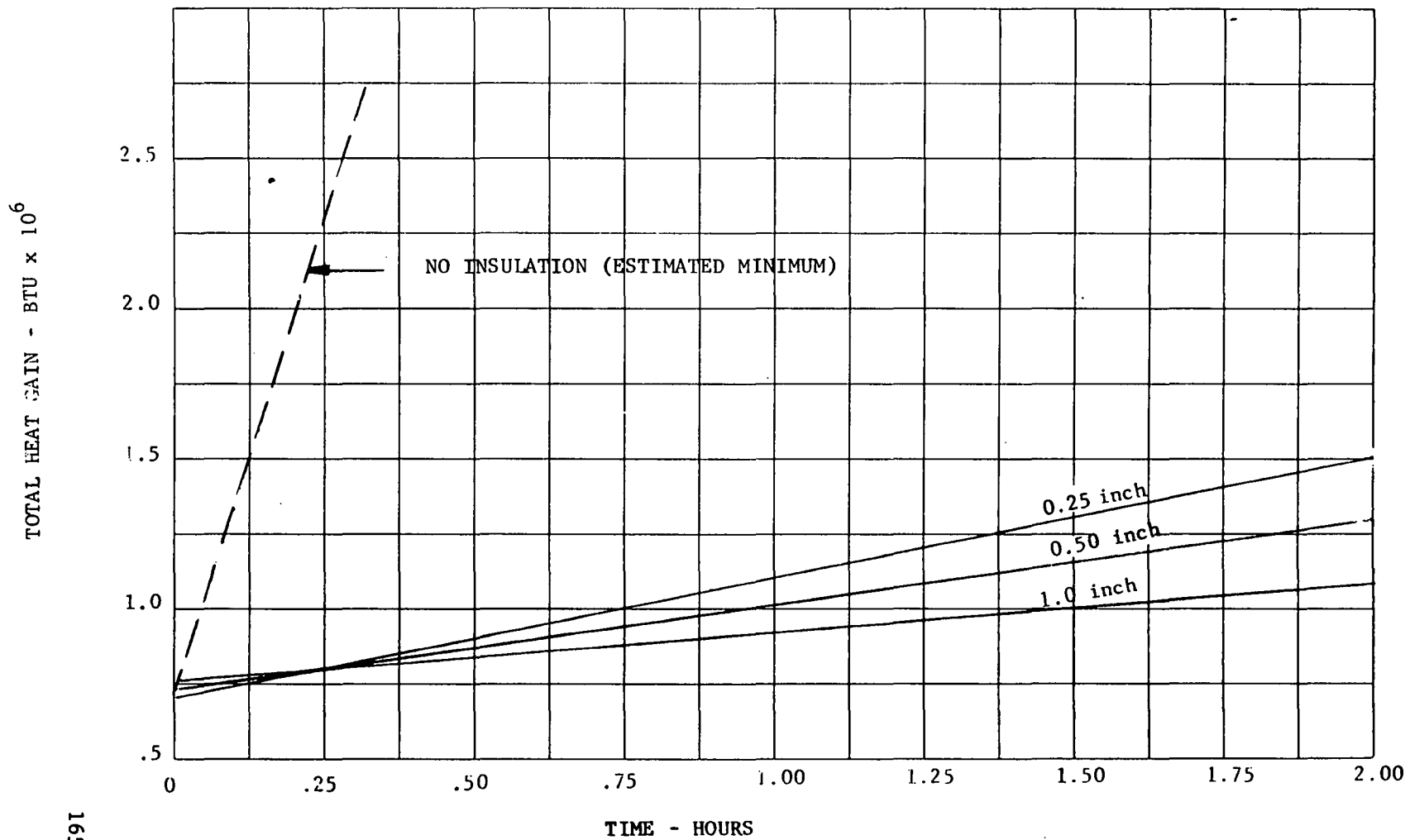


FIGURE D-2 TOTAL HEAT GAIN OF TRIPLE POINT LOX IN SSTO TANKS VS TIME FOR VARYING INSULATION THICKNESS

APPENDIX E - SYMBOLS

A	Area
Btu	British Thermal Unit
C	Specific Heat
°C	Degrees Centigrade
cm	Centimeters
\bar{F}	Radiation interchange factor
°F	Degrees Fahrenheit
ft	Feet
G	Gaseous
g_c	Gravitational acceleration
gal	Gallons
GLOW	Gross Lift-off Weight
gpm	Gallons Per Minute
H	Enthalpy
h	Convection Heat Transfer Coefficient
He	Helium
hr	Hour
H ₂	Hydrogen
in	Inch
j	joule
k	Thermal Conductivity
°K	Degrees Kelvin
Kg	Kilograms
Kj	Kilojoules
KPa	Kilopascal
KSC	Kennedy Space Center
L	Length
lb	Pound
LC	Launch Complex

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167

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LCH ₄	Liquid Methane
LH ₂	Liquid Hydrogen
LOX	Liquid Oxygen
L _f	Latent Heat of Fusion
m	meter, mass
min	Minute
MLI	Multi-layer Insulation
MPa	Mega Pascal
MR	Mixture ratio
N	Number of Layers
NBP	Normal Boiling Point
N ₂	Nitrogen
O ₂	Oxygen
P	Pressure
PPO	Polyphenylene Oxide
psi	Pounds Per Square Inch
psia	Pounds Per Square Inch Absolute
psid	Pounds Per Square Inch Differential
psig	Pounds Per Square Inch Gage
Q	Heat Input
• Q	Heat Leak
R	Thermal Resistance
°R	Degrees Rankin
sec	Second
SLCH ₄	Slush Liquid Methane
SLH ₂	Slush Liquid Hydrogen
SSTO	Single Stage to Orbit
T	Temperature
t	Time
TP	Triple Point
TSM	Tail Service Mast
V	Volume
• V	Volumetric Flowrate
W	Weight

w	Watts
x	Solid Content (Quality)
β	Coefficient of Volumetric Expansion
Δ	Differential
ϵ	emissivity
μ	viscosity
ρ	density
σ	Stefan-Boltzman Constant

SUBSCRIPTS

BL	Boundary Layer
e	Environmental
f	Friction
UB	Upper Boundary
UL	Lower Boundary
WL	Wall

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Study, Transmittal of Final Report

Ref: (a) Contract NAS3-21014, Reports of Work, Para. C

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Densification Study Final Report (1 Copy)

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